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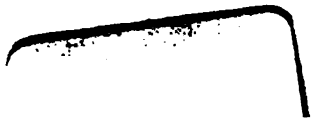

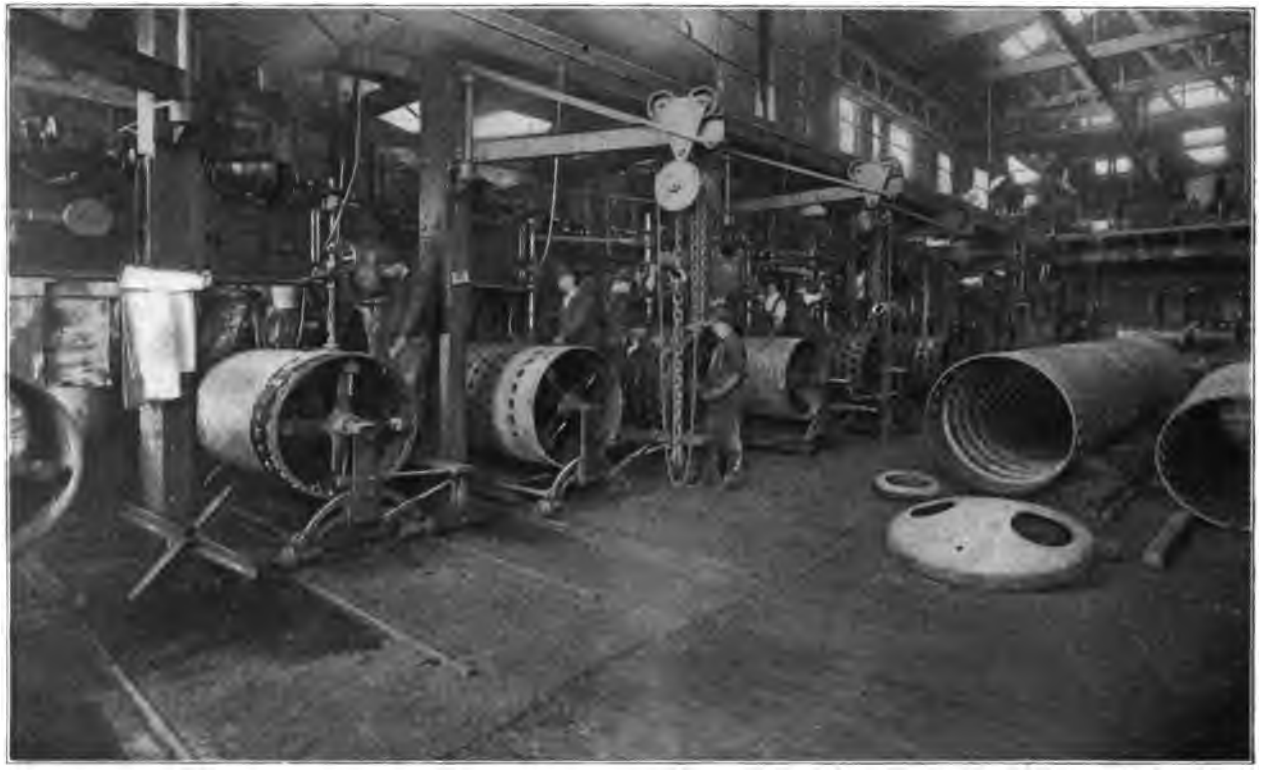

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MODERN ENGINEERING
STEAM-ELECTRICAL

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MACHINE DESIGN-BOILERS

AMERICAN SCHOOL of CORRESPONDENCE
at ARMOUR INSTITUTE of TECHNOLOGY
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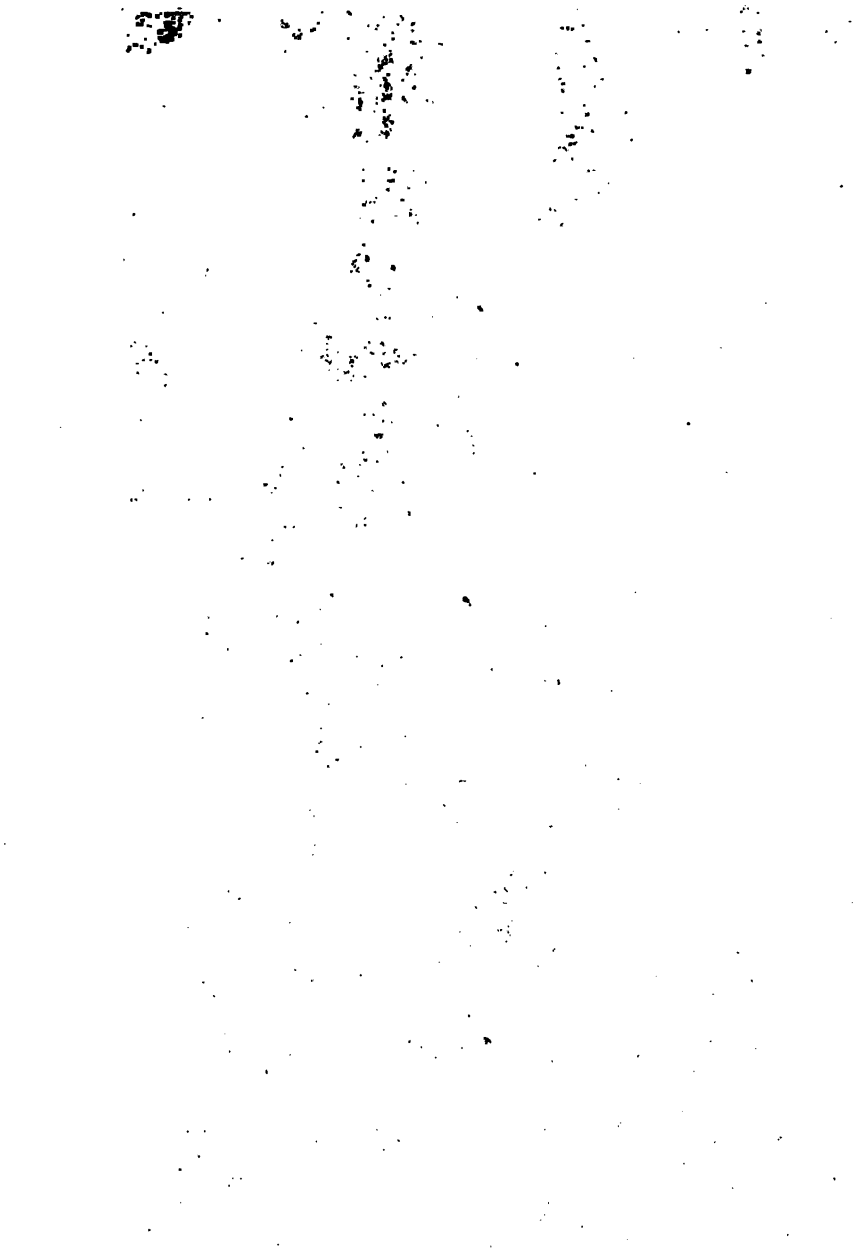

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Volume. IV.

MACHINE DESIGN BOILERS
FLYING MACHINE GAS ENGINES

SECTION I.

MACHINE DESIGN.

PART I.

Machine design treats of the design and construction of machines and their various parts.

A **machine** is a combination of movable parts, arranged on a supporting frame, and placed between the source of power and the work.

The *object* of a machine is to transform the energy supplied at the point where the machine receives its motion, into work at the point where the resistance is overcome.

The various parts may be arranged to *change the direction or velocity* of the power or to *overcome great resistance with small force.* For instance, a steam engine changes rectilinear to circular motion while a planer changes circular to rectilinear. A lathe is a familiar example of the change in velocity. Hydraulic presses and testing machines illustrate the overcoming of a great resistance by a small force.

A machine cannot move of itself nor create power. According to the law of the conservation of energy, no increase of power can be obtained from any machine. If a machine were frictionless, the product of the force exerted at the driving point and the velocity of the driving point would equal the product of the resistance and the distance through which the resistance is overcome in the same time.

The operation of machines depends upon two conditions; the *transmission* of certain forces and the *production* of definite resultant motions.

In designing machinery both of these conditions must be considered. The machine must be constructed so that each part will bear the strains placed on it and also have the proper relative motions.

The nature of these relative movements is independent both of the power transmitted and of the dimensions of the parts. The relative motions of the parts of a machine depend, not upon the absolute size of the parts, but upon their relative proportions; *i. e.*, the ratios of the number of teeth in the gears, the relative diameters of pulleys and the relative lengths of cranks or levers. Thus if a small model is built in strict proportion to the larger machine which it represents, its parts will have the same relative motions.

Pure Mechanism treats of the motion and form of parts of machines and the manner of guiding them.

Constructive Mechanism treats of the calculation of the forces acting on these parts. It involves the selection of materials, and the calculation of the dimensions for requisite strength and stiffness.

A **Mechanism** is a portion of a machine where two or more parts are so arranged and connected that the motion of one compels a definite motion of others.

Machines are made up of trains of mechanisms. The sewing machine, watch and printing press are examples.

Motion and **rest** are relative terms. If we consider some point or body as fixed, motion may be either relative or absolute. For this work the earth is assumed to be fixed and motion referred to it absolute.

A point moving in space follows a line, either curved or straight which is called its **path.**

Direction, like motion, is relative to continuous motion. If a point continues to move indefinitely in the same direction, it is said to have **continuous motion.** In this case, the path must be a closed curve. A shaft turning on its bearings or the crank-pin of an engine, is an example of this motion.

Intermittent Motion. When a part of a machine has definite periods of rest and definite periods of motion it is said to have intermittent motion.

Reciprocating Motion. If a point travels in the same path, alternately in opposite directions, its motion is said to be reciprocating. If the motion is reciprocating and also circular, it is called **vibration.**

Velocity. The ratio of the linear distance traversed to the time taken is called linear velocity, whether the path is a straight line or a curve.

Velocity may be uniform or variable according as the spaces traversed in equal times are equal or unequal.

$$\text{Velocity} = \frac{\text{Space}}{\text{Time}}$$

$$\text{Space} = \text{Time} \times \text{Velocity.}$$

$$\text{Time} = \frac{\text{Space}}{\text{Velocity}}$$

The unit of space is usually one foot; the unit of time one second. Hence velocity is expressed in feet per second.

If a point moves in a circular path, its velocity is a ratio of the length of its path traversed in a given time, to the time; or it is the length of the path through which the body moves in a unit of time. Suppose a point revolves in a circular path at a distance r from the center, and makes n revolutions in one second. The length of the path through which it moves in one revolution is equal to the circumference, and the length of the path it traverses in one second, or its *linear velocity*, is equal to the circumference multiplied by the number of revolutions.

$$\text{Circumference} = 2\pi r$$

$$\text{Linear Velocity} = 2\pi r n$$

A fly-wheel is 20 feet in diameter and revolves at the rate of 50 revolutions per minute. Find the linear velocity of a point on the periphery.

$$\text{Linear velocity} = 2\pi r n = 2 \times 3.1416 \times 10 \times 50$$
$$= 3141.6 \text{ feet per minute.}$$

How far from the centre of this wheel must a point be to have a velocity of 20 feet per second?

$$V = 2\pi r n, \text{ or } r = \frac{V}{2\pi n}.$$

$$\text{Then, } r = \frac{1200}{2 \times 3.1416 \times 50}$$
$$= 3.82 \text{ feet.} \quad \text{Ans.}$$

EXAMPLES FOR PRACTICE.

1. A wheel 3 feet in diameter makes 25 revolutions per minute. What is the linear velocity of a point on the circumference? Ans. 235.62 feet per minute

2. A wheel makes 300 revolutions per minute. How far from the center must a point be to have a linear velocity of 2827.44 feet per minute? Ans. $1\frac{1}{2}$ feet.

3. Find the linear velocity of a fly-wheel when the angular velocity is 188.5 revolutions per minute, the wheel being 10 feet in diameter. Ans. 942.5 feet per minute.

Revolution. A point revolves about an axis when it describes a circle the center of which lies within, and its plane of rotation is perpendicular to, the axis. If the axis passes through the body, as in the case of a wheel, the motion is called both rotation and revolution.

A body, like the earth for instance, may rotate about its own axis and also revolve in an orbit about another axis.

Cycle of Motions. In case the parts of a mechanism go through a series of motions which are repeated, each time the order of the motion of the several parts being the same, the series is called a cycle of motions. In the steam engine every revolution is a cycle because each series of motions is repeated for every revolution. Two revolutions of the crank are necessary for one cycle in some types of the gas engine.

The part or piece of mechanism which causes motion, or to which the power is applied is called the **driver** and the part whose motion is caused by the movement of the driver is the **driven** or **follower.**

Frame. The structure which supports or holds in position the moving parts and regulates the path of motion is called the frame. The motions are often referred to the frame, as it is usually fixed, that is, without motion. An exception to this is the locomotive frame.

Transmission. One object cannot move another unless the two are in contact or are connected by some body that is capable of communicating motion from one to the other. The above state-

ment does not take into account the action of natural forces, such as gravity, magnetism, etc.

Motion transmitted by contact is seen in friction gearing, gear wheels, etc. Belts, rope gearing, levers, links, etc., are examples of motion by intermediate connection. Connectors are either rigid or flexible.

Motion. Mechanism may be used to change the motion of the follower from that of the driver. It may differ in direction, kind, or velocity.

Work is the overcoming of resistance through distance. It is the product of the resistance and the space through which it is overcome. If a body is lifted from the earth, against the attraction of gravity, the resistance is the weight of the body and the distance is the height to which the body is raised. The work done is equal to the weight of the body multiplied by the distance.

The **Unit of Work** is the foot-pound, that is, the amount of work done in lifting one pound through one foot. If F equals the force or weight and S equals the space or distance through which F is moved, then work $= F \times S$. S = velocity multiplied by time $= V \times T$. Then if we raise 5 pounds to a height of 20 feet we do $5 \times 20 = 100$ foot-pounds of work.

Energy is the capacity for doing work. There are two kinds of energy, potential and kinetic.

Potential Energy is possessed by a body by virtue of its position or state. It is measured in foot-pounds. A weight suspended at an elevation has potential energy equal to the product of its weight multiplied by its height above the ground, for if allowed to descend it has power to do this amount of work. As an example of energy possessed by a body by virtue of its state, may be mentioned the energy in fuel, which possesses power to do work in combustion.

Kinetic Energy in a moving body is the work which it is capable of performing against resistance before it is brought to rest. It is equal to the work required to bring it from rest to its actual velocity. Kinetic energy is the product of the weight and the height through which it must fall to acquire the actual velocity. From the laws of falling bodies this height

equals the square of the velocity divided by twice the value of the earth's attraction. Then

$$h = \frac{v^2}{2g}$$

and energy,

$$E = wh = \frac{wv^2}{2g}.$$

The **weight** of a body divided by g is called the **mass**, *i. e.*,

$$m = \frac{w}{g},$$

then substituting m for $\frac{w}{g}$ in the equation

$$E = \frac{wv^2}{2g}, \text{ it becomes } E = \frac{mv^2}{2}.$$

Energy is the capacity of doing work. The units of work and energy are the same; then

$$F\,S = wh = \frac{wv^2}{2g} = \frac{mv^2}{2}.$$

Power is the *rate* of performing work. It is equal to the work done divided by the time and is expressed as foot-pounds per minute or per second. Thus **horse-power** is a measure of power, being equal to 33,000 foot-pounds per minute or 550 foot-pounds per second.

$$\text{Power} = \frac{FS}{T}.$$

MATERIALS.

The principal materials used in the construction of machinery are cast and wrought iron, copper, wood, brass and other alloys. The properties and processes of manufacture for iron and steel have been described in Metallurgy.

CAST IRON.

Cast iron is used to a considerable extent in the construction of machines. For the heavy massive parts, the frames of lathes, steam-engines, planers, etc., for example, it is the best material. It is *not* suitable for parts requiring strength, elasticity or those

subjected to shocks. For this reason piston-rods, connecting-rods, shafts, etc., are usually made of steel or wrought iron.

Many complicated shapes that cannot be forged are readily cast. The ease with which parts may be given the desired shape makes cast iron valuable.

Cast iron contains 3 to $4\frac{1}{2}$ per cent. of carbon with a little silicon. The hard and white varieties are used in the manufacture of wrought iron. The gray irons are used in the foundry.

Cast iron is made into the desired forms by melting it in a cupola and pouring into moulds. The moulds are made in sand or loam from patterns of pine wood. Patterns are made a little larger than the required casting because iron in solidifying contracts about $\frac{1}{8}$ inch per foot in each direction. This contraction is called shrinkage. In making a pattern a shrinkage rule is used which is about $\frac{1}{8}$ inch longer per foot than the standard.

Castings are likely to be put into a state of internal stress because of contraction when cooling. If some parts of the casting contract more than others, the casting may become twisted. Thin parts of the castings solidify first. The contraction of the fluid parts strains the portions already set and their resistance to deformation causes stresses to be set up in the parts which are solidifying.

For example, the form shown in Fig. 1 has a rigid flange

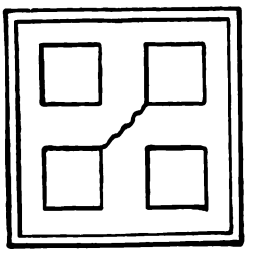


Fig. 1.

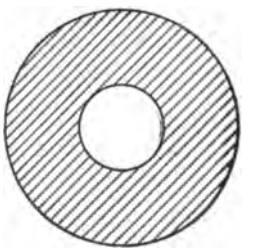


Fig. 2.

surrounding the inner part. If the contraction of the cross piece takes place more slowly than the rim, it is likely to fracture. In a thick cylinder, as shown in section in Fig. 2, the outer portions solidify and begin the contraction. The contraction of the inner induces pressure in the outer portion, which being rigid causes a resistance to contraction of the inner layers and puts them in ten-

sion. A cylinder so constructed is not strong to resist bursting pressure. If the cylinder is cast while water circulates through the core, the reverse distribution of initial strains is set up. This insures a stronger cylinder because the inner layers are in a state of compression and the outer portions are in tension.

The arms of pulleys may be broken by tension if the rim is thin and rigid. If the arms set first the rim may break near them. To have successful castings, the designer must carefully consider the dimensions of the various parts.

On account of these initial strains, that cannot be calculated, cast iron is unreliable. Cast iron structures usually have excessive dimensions to insure safety.

In cooling, the crystals of cast iron arrange themselves per-

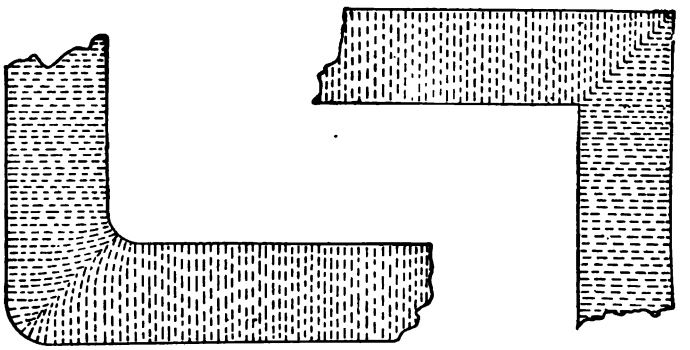


Fig. 3.

pendicularly to the surfaces from which heat radiates. For this reason all corners should be well rounded as shown in Fig. 3, so that the arrangement of the crystals will make the castings strong.

Chilled Castings. If castings are cooled rapidly during solidification, the graphite is prevented from separating from the iron. This causes the iron to become harder. In order to chill the cast iron, the mould is made of or lined with this same material. The mould which is lined with loam for protection, is a good conductor of heat and the molten cast iron is cooled or chilled during solidification.

The chilling usually extends to a depth of $\frac{1}{8}$ to $\frac{3}{8}$ of an inch from the surface; the interior remaining soft.

Malleable Cast Iron. Malleable cast iron is made by surrounding castings with oxide of iron, powdered red hematite or peroxide of manganese; keeping them at a high temperature for a considerable time according to the size of the casting. The elimination of carbon converts the cast iron into a crude form of wrought iron. Malleable castings will stand blows better than ordinary castings.

Cast iron is stronger than wrought iron when under pressure; but it is much weaker under tension and impact.

WROUGHT IRON.

Wrought iron is made from cast iron by eliminating part of the carbon. It is strong and tough and can easily be welded. For these reasons it is used for parts of machines and structures requiring strength and of simple form. Wrought iron parts are shaped by forging and finished in the machine shop; steam hammers being used on the heavy portions.

Wrought iron is rolled into plates, round and square bars, angle, tee, channel, I beam sections, etc. Large wrought iron structures are built up of bars or plates riveted or bolted together.

Wrought iron that has been rolled when cold has a greater tensile strength than before rolling; but its ductility and toughness is reduced. Annealing, or heating the iron to a red heat and allowing it to cool slowly, restores it to the original condition.

Compression of iron when cold increases its strength but reduces its ductility and toughness; annealing reduces strength and increases toughness and ductility. If the iron is rolled or hammered when hot, compression and annealing are carried on at the same time.

Wrought iron is used for piston-rods, shafts, connecting-rods, bolts, nuts, chains, etc.

STEEL.

Steel is far the most useful material used in machine design. Its properties depend upon the percentage of combined carbon, silicon, manganese, phosphorus, etc. Steel contains from .15 to 1.5 per cent. of carbon.

Formerly it was difficult to get sound castings, but by the use of silicon, aluminum and other elements and prolonged annealing, the internal stresses are destroyed.

Steel can be welded, but greater care is necessary than in the welding of wrought iron.

Tempering greatly increases the usefulness of steel, since it becomes hard if heated and cooled suddenly. With good steel almost any desired hardness may be obtained. The steel is heated to the temperature indicated by the color of the oxide which forms at its surface and is then quenched in oil or water. Hardness makes it suitable for cutting tools. When tempered it is hard, strong, has high elastic limit and little ductility.

COPPER.

Copper is a reddish metal of great ductility and malleability. It is usually rolled or hammered into shape because it doesn't cast well. Copper can be welded, but as it requires considerable care to make a good joint, pieces are more often joined by brazing. It can be drawn into wire. The tensile strength of cast copper is about 20,000 pounds per square inch; of forged copper about 30,000 pounds per square inch.

Hammering, rolling and wire-drawing increases the tensile strength, but makes it hard and brittle. It can be made soft and tough by annealing. It is expensive and is used for wire, fittings and tubing. Its strength is less than that of wrought iron and decreases rapidly with rise of temperature.

ALUMINUM.

Aluminum is a soft, ductile, malleable metal of bluish white color. It is very light; next to magnesium the lightest of the useful metals. Its strength is about one-third that of wrought iron. Aluminum casts well, the shrinkage being about the same as brass. The readiness with which aluminum unites with other metals makes it valuable for alloys. It can be electrically welded but doesn't solder well.

BRONZE.

Bronze, or gun-metal, is an alloy of copper and tin; about 90 parts copper and 10 parts tin. It makes good castings. Bronze is harder and less malleable than copper. Copper-tin alloys are used for bearings because it is softer and wears faster than wrought iron or steel shafts.

The hardness of bronze depends upon the proportion of tin; to increase hardness increase the amount of tin. An alloy of 92 parts copper and 3 parts tin is a soft bronze used for gear wheels.

Phosphor-bronze is made by mixing 2 or 3 per cent. of phosphorus with ordinary bronze. Soft phosphor-bronze has a tensile strength of about 45,000 pounds per square inch; harder varieties have about 65,000 pounds and hard unannealed wire has about 150,000 pounds. It is used for pump-rods, propellor-blades, etc.

Manganese bronze, called white bronze, is an alloy of ordinary bronze and ferro-manganese. Like phosphor-bronze it is used in marine work, because it resists the corroding action of sea-water. Manganese bronze is equal in tensile strength and toughness to mild steel and can be easily forged.

BRASS.

The alloy of copper and zinc is called brass; sometimes tin and a little lead are added. For bearings it has about 60 per cent. copper, 10 per cent. zinc and 30 per cent. tin and lead. Naval brass has 62 per cent. copper, 1 per cent. tin and 37 per cent. zinc. Red brass consists of about 37 per cent. copper and for the rest about equal parts of tin, zinc and lead. Brass is used for bearings, wire, fittings and ornamental work. Its tensile strength is about 23,000 pounds per square inch.

FUSIBLE ALLOYS.

Fusible alloys are made of tin, lead and bismuth. The melting point varies with the percentages of the various constituents. If made of 2 parts lead and 1 part tin, it melts at 475° F.; if 1 part lead, 1 part tin and 4 parts bismuth, the melting point is about 200° F. An alloy of 1 part cadmium, 4 parts bismuth, 1 part tin and 2 parts lead melts at 165° F.

BEARING ALLOYS.

The principal constituents of bearing alloys are copper, tin, lead, zinc, antimony and aluminum. The bronzes contain a large per cent. of copper. A good bearing alloy is made of copper, 77 parts by weight, tin 3 parts and lead 15 parts.

Babbit metals have various proportions; hard babbit having about 89 per cent. tin, 4 per cent. copper and 7 per cent. antimony.

There are many other alloys containing the metals in varying proportions according to the intended use.

WOOD.

Wood is but little used in machine construction. Soft woods like pine are used for patterns; hard varieties, oak and lignum-vitæ for examples, are used for bearings. Sometimes levers are made of wood and the pulleys of some lathes are constructed of the same material. The cogs of mortise wheels are often made of beech or horn-beam.

SHOP PROCESSES.

In designing machinery it is necessary that the parts may be easily made. A finely finished pattern is of no value if it cannot be taken from the mould. Complicated castings and forgings should be used only when absolutely necessary. Simple designs are usually the best.

For **casting**, patterns or models are made from wood in the pattern shop. The pattern maker has to consider and make allowance for shrinkage in casting, for turning, boring and finishing. He arranges the patterns in such manner that the moulding, casting and finishing may be most cheaply done. Some parts can be moulded only by the use of cores. Parts to be finished by cutting tools must be so placed that they will not be unsound by reason of blow holes. The founder follows the specifications of the drawings; mixing the pig iron in different proportions so as to get the required strength and softness.

Forging is the operation of shaping wrought iron or steel without melting. These materials become plastic without fusing. The pieces are hammered or rolled into shape. If the work is

light the smithing is done by hand; but when large forgings are made, steam hammers are used. When the pieces are very large, great skill is required to arrange the operation so that the result shall be a homogeneous sound piece.

Fitting, finishing, boring and **turning** are the operations of cutting the rough products of the foundry and forge to accurate dimensions. Fitting, boring and turning are done by steel cutting tools which shape the metal when cold. Cutting operations include chipping and filling, drilling, turning, planing, shaping and milling.

Conical surfaces, screws and nuts can be made in the lathe.

STRAINS IN MACHINES.

There are forces acting on the several parts of a machine which will cause them to give way if they are not sufficiently strong. Among these forces are the following:

1. The **useful load** caused by the power transmitted from the point of receiving the energy to the point where the useful work is accomplished.
2. **Resistance** due to friction in the machine.
3. Forces due to **inertia** caused by change of velocity of the moving parts.
4. **Weight** of parts of machines.
5. **Centrifugal forces** caused by changes in direction of motion.

The total action caused by the above forces is called the straining action on whatever part is considered. This straining action varies with the changes of working load, with the variation of position of the parts, with the change in speed, etc. In designing the various parts it is necessary to consider under what conditions the straining actions are greatest and calculate the dimensions of those parts to safely stand that action.

It is obvious that the maximum working load must be *less* than the breaking load. In most cases it should be very much less. Generally it is much easier to determine (by means of testing machines) the breaking strength than it is the working stress. In order to be sure of sufficient strength, it is customary to divide the breaking strength by the **factor of safety**, to find the allowable

working load. Results from actual cases provide us with average factors of safety for various conditions. In case the straining actions are well known and the stresses are steady, the factors are small. A large factor is necessary when the straining actions are likely to be greatly in excess of the calculations, when the material is not reliable and when the parts are liable to shock. Some designers never use the term factor of safety, but know from experience that the various materials will safely carry a certain load under given conditions.

In most cases a permanent set would be injurious; it might prevent the movements of some parts of a machine. Under these conditions it is evident that the working stress must be less than the elastic limit.

MACHINE DRAWINGS.

Machines are designed from principles obtained by successful practice and from mathematical calculations. In order that both the designer and the mechanic may have a clear idea of the work, the designer makes a **drawing** of the machine. The drawing indicates the size and shape of the various parts and how they are to be put together. By means of the drawing, the designer calculates the relative motions of the parts and arranges them so that they will not interfere with each other. He calculates the sizes for strength and considers the modifications which will produce the greatest efficiency, or least cost of manufacture. The drawing indicates how the work is to be performed and distributed in the different shops. All dimensions, names of materials and finish marks should be clearly shown so that the workman, by carrying out accurately the ideas of the designer, may produce the desired machine.

Usually several views of the part to be made are shown. Sometimes it is necessary to show **sections** in order that the internal construction or sectional shape may be easily understood. These sections are usually drawn through the axis, or center, but it is sometimes advisable to show sections of other portions. Where the drawing shows a section, the portions of metal or wood supposed to be cut are covered with parallel lines at equal distances and usually oblique. These sections are called **hatched, cross hatched,** or simply **sectioned.** The character of the lines

full lines, dotted, broken, light or heavy, indicate the material supposed to be cut. One kind indicates cast iron, another steel, another brass, etc. There is no standard for cross hatching, different draughtsmen using lines of various character. There is likely to be a confusion unless the parts have the name of the material printed on or near it, or a key is provided.

Fig. 4 shows the lines as generally used; those representing

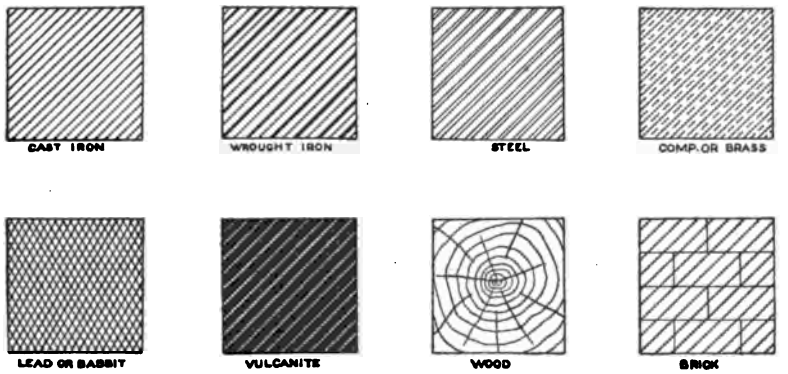


Fig. 4.

cast iron, brass, wood and lead being almost universal, the others are subject to more change. The lines may run from left to right, or right to left; in case two or more parts of the same metal are brought together it is necessary to avoid confusion by varying the direction and angles of the lines. If the hatching were to be the same, the parting line would be confused and one might think it all one piece.

When drawing designs of the details, it is well to make them as large as convenient. The scales in general use are full size, half size, 3 inches or $1\frac{1}{2}$ inches $=$ 1 foot. A drawing is never made by such scales as 2 inches or 4 inches $=$ 1 foot.

A working drawing is one that shows all the dimensions of an object in such a manner that it may be made by reference to the drawing. Usually three views are sufficient; two elevations, taken at right angles to each other, and a horizontal projection or plan. Often the plan may be dispensed with if the object represented is of simple form. Sections to show the interior construction are sometimes added.

It is not sufficient to draw the various views and sections the correct size; the **dimensions** also should be placed on the drawing. The workman can tell immediately the size of any given part without scaling it from the drawing. Although desirable, it is not necessary that a drawing be made accurately, provided all the dimensions are put in correctly. The chances of error are greatly reduced by removing the necessity to scale off dimensions.

Dimensions should be used systematically and wherever necessary. In placing dimensions on drawings, a line should be drawn from one point to the other. The number representing the dimension is placed in the space left for it at the centre of the line. These lines should be either fine full lines or dashes about $\frac{1}{2}$ inch long. Arrow heads are placed on the ends of the line, the heads or vertex of the arrow just touching the points or lines. In case the dimensions are very small, the arrow heads may be outside instead of between the lines, or pointing toward each other. The dimensions should be written in feet, inches, halves, quarters, eighths, sixteenths, etc. of inches. Fractions should be reduced to lowest terms. Write $\frac{3}{8}$, not $\frac{6}{16}$, nor $\frac{12}{32}$. The dividing line of the fraction should be parallel to the direction of the dimension line; never an oblique line because the oblique line may be mistaken for some part of a number. Feet are represented by the symbol $'$, inches by $''$. The inch marks should be placed after the fraction not between the whole number and the fraction; thus, eight feet, seven and three quarters inches should be written $8' - 7\frac{3}{4}''$, not $8'\ 7''\frac{3}{4}$. When the length is even feet it is usual to write it $8' - 0''$ in order that the workman may know that the inches were not left off by mistake.

Figures showing dimensions should *always* read *along* the line whose length they show, and whenever possible they should be arranged to read from either the bottom or the right-hand side of the sheet.

It is necessary to get in all the important dimensions, especially the "over-all" dimensions, so that the workman will not be compelled to add up several small dimensions in order to select his stock.

Dimensions are often placed between two views and usually outside the several views. When placed outside, extension lines

are used, that is, a fine or dash line is drawn as a continuation of lines or edges.

In placing the dimensions of a circle, give the diameter, not the radius. When an arc is used, give the radius. Holes may be located by giving the dimensions from the outside, or from the center of figure, to the center of the hole. The distance from center to center shows their distance apart. In case the holes are arranged in a circle, as in a cylinder head for instance, give the diameter of the circle whose circumference passes through the center of the holes.

In making sectional views the plane of the section passes through the center line of a shaft, bolt, screw, or cylinder, and the cylinder part is not represented in section but in full.

It is usual when making mechanical or working drawings to do the work in pencil first, and then ink in the necessary lines, or a piece of tracing cloth is placed over the pencil drawing and the lines which show through are then inked. The latter method is used in case a number of blue prints are desired for the shops or office.

In the pencil work, **accuracy** is necessary. Some beginners think they can correct inaccuracies in pencil by care in inking. The lines should be located exactly of the required length. A hard pencil; 4 H or 6 H is generally used. A hard pencil, when sharp, makes a depression in the paper which cannot be erased. For this reason press lightly on the pencil.

In inking, it is better to make circles, arcs of circles and curved lines *first*. It is much easier to make straight lines meet arcs, or to make them tangent to circles or arcs, than the reverse. To indicate an edge or intersection of two planes a full line ——— is used; edges or intersections which are concealed are represented by dotted lines . . . . . Dot and dash lines, — . — . — . — . or — . . — . . — . . — . . indicate center lines or axes. A fine line or a series of long dashes —— —— —— —— is used for dimension lines. Titles, various views, sections, names of materials, etc., are **lettered** on the drawings. The styles and the care taken in this work varies with the draughtsman or with the amount of time at his disposal. In every case all lettering should be neatly done and of some clear cut simple form.

Marks indicating in which shop the work is to be done and for classification are also placed on drawings. Fig. 5 shows a working drawing of a three inch pillow block which illustrates the above principles. The three views side, plan and end are half in section.

DESIGN.

In this book and in Machine Design II, the dimensions of some of the drawings are given in letters instead of figures, so that the relation between the different dimensions may be explained. In general, however, this should not be done. The dimensions placed upon a drawing should be in figures, for letters without an explanatory table convey no idea of the size.

After the proportions of the parts of a machine have been properly determined to give the required motions, the length, breadth and thickness of the parts must be determined. The dimensions of some parts depend upon the strength required, and should be calculated after a careful consideration of the forces acting and the greatest allowable stress. Surfaces which slide upon one another, such as collars and bearings, should be arranged to provide sufficient surface to distribute the friction and prevent heating. Sometimes the thickness necessary for requisite strength may be very small, yet the part must be made much thicker in order to secure a sound casting. Many good designers never use bolts or pins of less than a certain diameter, no matter how small the load may be. Smaller bolts and pins might easily be mislaid, or broken when being put into place.

Some parts of machines, bolts, nuts, screws, pipes, etc., can be obtained in standard sizes from various factories. The designer should know these standard sizes and make his details of such shapes that they will conform. The designer must also keep in mind the processes and tools to be used so that the construction will not be too difficult or expensive.

Usually dimensions are expressed in feet and inches and such fractions of inches as $\frac{1}{2}$, $\frac{1}{4}$, $\frac{1}{8}$, $\frac{1}{16}$, $\frac{1}{32}$. $\frac{1}{3}$, $\frac{1}{5}$, $\frac{1}{6}$ or $\frac{1}{10}$ are never used since the scales in shops are not divided in these fractions. Decimals are used only for great accuracy or in the design of gear teeth.

All machines are made up of different combinations of *simple* principles. The designer must know these principles and the relations they bear to one another. He must also have a thorough

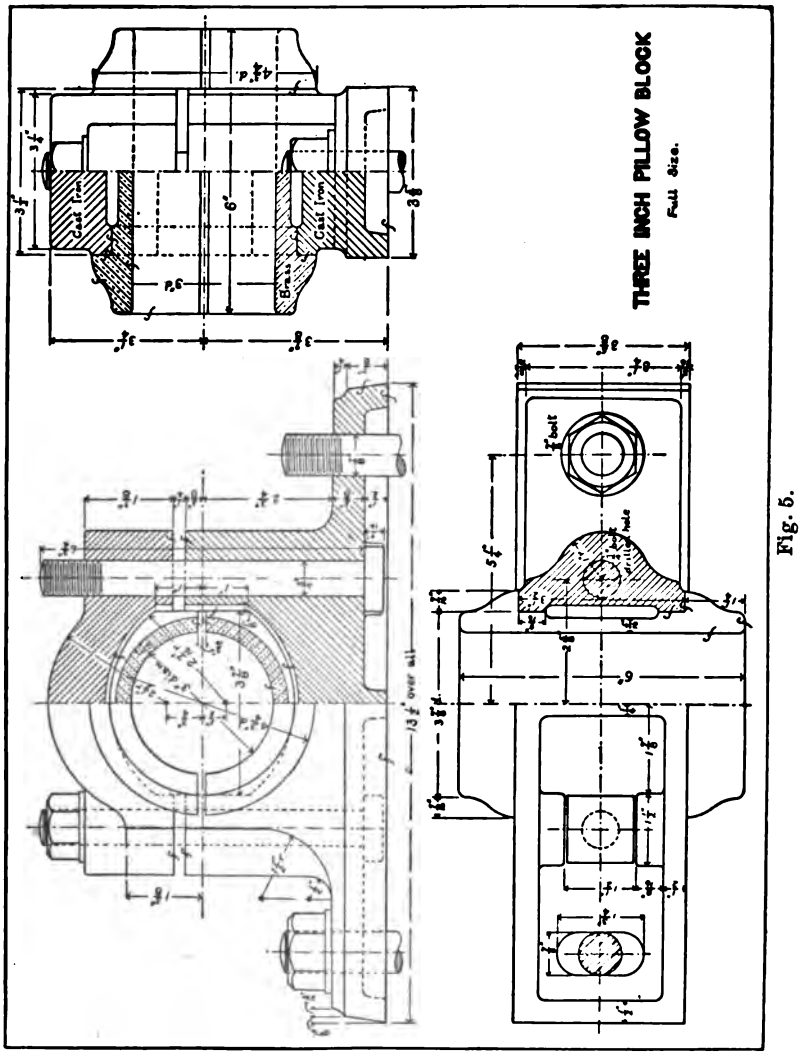


Fig. 5.

knowledge of the machine; the work it is to do; the character of power to be applied and in some cases the location and surroundings. A study of machines that have been designed to do similar work will be of great assistance. A wide knowledge of machines

permits the designer to have many ideas of details for the new machine.

Often several complete drawings are necessary before the final result is attained. An idea that seems desirable may not prove so when the details are worked out. Sometimes the relative motions are all right, but when the parts are designed for strength, they are found to interfere with each other. The expense of construction or the difficulties of manufacture may render a good design impracticable.

All notes, calculations and sketches of details and combinations should be carefully preserved for reference. Ideas worthless for some particular machine may be found very valuable for another. It is well to keep sketches, calculations and memoranda in books rather than on loose sheets of paper that may become lost or misplaced.

The estimates for *weights* and *cost* of machinery are made from drawings. The volume is found by mensuration and when multiplied by the weight of a cubic unit of the material gives the weight. Considerable skill and experience is sometimes necessary to estimate the volumes of irregular shaped parts. If the weights are known the cost is estimated from market values. The *time* necessary for completion is also judged from the amount of work and the ease with which it can be accomplished.

At the beginning an inexperienced designer is usually taught by making drawings of details which have been designed by others. Often he is employed for some time simply tracing drawings. After a little he is given the principal dimensions of the simpler parts and instructed to make working drawings for the pattern, forge or machine shops. During this time he becomes familiar with methods adopted in shops, standard dimensions, allowable stresses for the various materials, methods of adjusting wear and lubrication, necessary dimensions for sound castings, etc.

The following are a few practical rules or suggestions that are unconsciously kept in mind by the successful engineer, but are often forgotten by the inexperienced.

Means for **adjusting** all parts subject to wear, should be provided.

Make the motion of all parts *positive* if possible; avoid the use of springs and weights for producing motion.

Provide means for **lubrication** wherever necessary.

Construct the parts that may wear or break so that they will be **accessible** for adjustment or repairs.

Cranks, belts, levers and gear wheels are preferable to cams, screws and worm-wheels.

Avoid the use of **tap bolts** and **studs**; use through bolts or T head bolts if possible.

If convenient make the pressure per square inch on slides small.

FASTENINGS.

Bolts, Nuts, Keys, Cotters, etc.

A **screw** is a cylindrical bar, upon the surface of which a helical projection called the thread has been formed. A cylindrical helix is the curve generated by the revolution of a point about the surface of a cylinder, while moving along the axis at a constant rate.

A **nut** is a short hollow prism, upon the inside of which are formed grooves which correspond accurately to the threads of the screw.

The screw and nut when used for fastening is called a bolt. Screws are also used to transmit motion and to adjust the relative positions of two pieces. Screw threads are usually triangular or square in section. The Whitworth and U. S. standard threads are triangular. Square threads are used chiefly to transmit motion. There is less friction and less wear than with triangular threads, but they are more expensive.

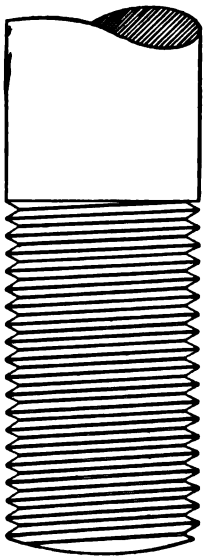


Fig. 6.

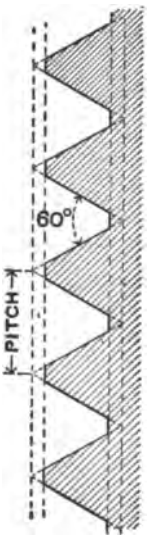


Fig. 7.

The Standard Thread, Fig. 6, shows the American or Sellars triangular thread. The construction is shown enlarged by Fig. 7.

The sides of the thread are inclined at an angle of 60°. The section of the thread is an equilateral triangle having its top cut off so that the flat portion is $\frac{1}{8}$ of the pitch in width. The angles at the bottom are filled in similarly. The real depth of the thread is a little less than the altitude p of the triangle; it is .65 p.

Sharp V threads are those cut without the flat top and bottom; the section being an equilateral triangle.

The **pitch** of a screw is the distance the screw advances during one revolution; it is the distance from one thread to the next. Another method of indicating the pitch is to give the **number of threads per inch.** For instance we say a screw has 11 threads per inch; that is, it will advance one inch during 11 revolutions or the pitch is $\frac{1}{11}$ of an inch, or .091 inch.

The pitch of threads depends upon the diameter of the bolt. The following equations give approximately the pitch, and the diameter at the bottom of thread in the U. S. standard.

$$p = .24\sqrt{d + .625} - .175 \text{ inch},$$
$$d_1 = d - 1.3p = d - 2p_1$$

In these formulas p = pitch, d = diameter of bolt, d_1 = diameter at the bottom of the thread, and p_1 = real depth of the thread.

Let n = the number of threads per inch; then

$$n = \frac{1}{p} \text{ and } d_1 = d - \frac{1.3}{n}$$

The external diameter of a bolt is $1\frac{7}{8}$ inches, to find the pitch, the number of threads per inch, the diameter at root of threads and the depth of thread we proceed as follows:

$$p = .24\sqrt{1.875 + .625} - .175 = .204 \text{ inch.}$$
$$n = \frac{1}{.204} = 5 \text{ (about) which makes the pitch .20.}$$
$$d_1 = d - 2p_1$$
$$2p_1 = d - d_1$$
$$p_1 = \frac{d - d_1}{2} = \frac{.26}{2} = .13$$

Then the pitch is $\frac{1}{5}$ or .2 inch: there are 5 threads per inch, diameter at root of threads is 1.615 inches and the depth of thread equals .13 inch.

The following table gives the principal dimensions of U. S. standard or Sellers threads.

Diameter of Bolt (Inches)	Number of Threads (Per Inch.)	Diameter at Bottom of Thread. (Inches.)	Area at Bottom of Thread (Square Inches.)
$\frac{1}{4}$	20	.185	.0269
$\frac{5}{16}$	18	.240	.0452
$\frac{3}{8}$	16	.294	.0679
$\frac{7}{16}$	14	.345	.0935
$\frac{1}{2}$	13	.400	.1257
$\frac{9}{16}$	12	.454	.1619
$\frac{5}{8}$	11	.507	.2018
$\frac{3}{4}$	10	.620	.3019
$\frac{7}{8}$	9	.731	.4197
1	8	.838	.5515
$1\frac{1}{8}$	7	.939	.6925
$1\frac{1}{4}$	7	1.064	.8892
$1\frac{3}{8}$	6	1.158	1.0532
$1\frac{1}{2}$	6	1.283	1.2928
$1\frac{5}{8}$	$5\frac{1}{2}$	1.389	1.5153
$1\frac{3}{4}$	5	1.490	1.7437
$1\frac{7}{8}$	5	1.615	2.0485
2	$4\frac{1}{2}$	1.711	2.2993
$2\frac{1}{4}$	$4\frac{1}{2}$	1.961	3.0203
$2\frac{1}{2}$	4	2.175	3.7154
$2\frac{3}{4}$	4	2.425	4.6186
3	$3\frac{1}{2}$	2.629	5.4284
$3\frac{1}{4}$	$3\frac{1}{2}$	2.879	6.5099
$3\frac{1}{2}$	$3\frac{1}{4}$	3.100	7.5477
$3\frac{3}{4}$	3	3.317	8.6414
4	3	3.567	9.9930
$4\frac{1}{4}$	$2\frac{7}{8}$	3.798	11.3292
$4\frac{1}{2}$	$2\frac{3}{4}$	4.027	12.7366
$4\frac{3}{4}$	$2\frac{5}{8}$	4.255	14.2197
5	$2\frac{1}{2}$	4.480	15.7683
$5\frac{1}{4}$	$2\frac{1}{2}$	4.730	17.5717
$5\frac{1}{2}$	$2\frac{3}{8}$	4.953	19.2676
$5\frac{3}{4}$	$2\frac{3}{8}$	5.203	21.2617
6	$2\frac{1}{4}$	5.423	23.0978

The **Whitworth** triangular thread shown at D, Fig. **10**, is used in England. The angle between the surfaces is 55°, and $\frac{1}{6}$ of the depth or altitude of the triangle is rounded off instead of

being flat as in the Sellers thread. The pitch and diameter at the bottom of the thread is found as follows:

$$p = .08d + .04,$$

$$d_1 = d - \frac{1.28}{n} = .9d - .05.$$

A screw with a **square** thread is shown in Fig. 8 and an enlarged section of the thread in Fig. 9. The pitch of the square thread is usually double that of the triangular or V thread for a bolt or screw of the same diameter. The pitch is about $\frac{1}{6}$ the

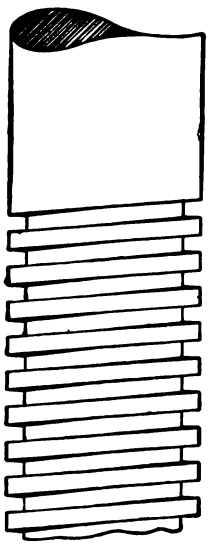


Fig. 8.

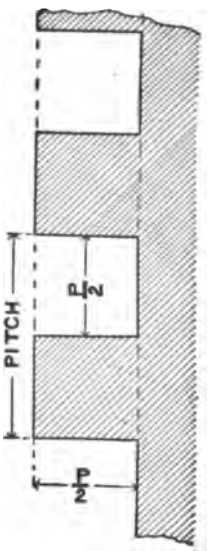


Fig. 9.

diameter of the screw; the diameter at the bottom of the thread is $\frac{4}{5}$ the diameter of the screw and the depth of thread $\frac{1}{2}$ the pitch or $\frac{1}{10}$ the diameter of the bolt. The edges of the square thread are slightly rounded to prevent flattening and to prevent binding of the nut.

If the thread is to be subjected to very rough usage the rounding is carried further so that the section of the thread is like

that shown at A, Fig. 10. The thread shown at B is like a square thread, but instead of having a square section the thread **tapers** from the root to the point. This taper is given to the thread because it is easier to cut than a square thread. This form is often used as lead screws for lathes. The taper allows the nut, which is in two parts, to engage and disengage easily. The

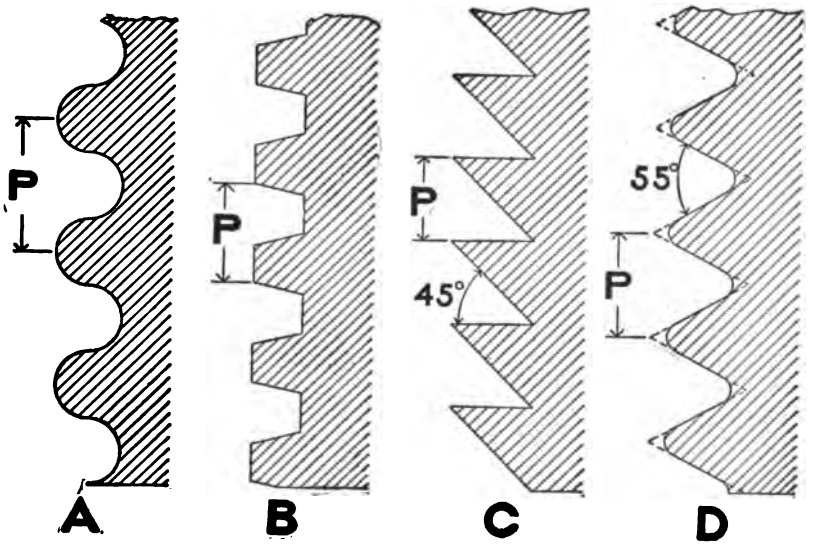


Fig. 10.

trapezoidal or buttress thread is shown at C. It is used for transmitting motion or when the force acts always in the same direction. One face is normal to the axis of the screw and the other is inclined at an angle of 45 degrees. The top is usually cut off and the bottom filled in as in the U. S. standard thread; the amount being about the same, that is, about $\frac{1}{8}$ the depth. The pitch is equal to the theoretical depth since the angle of the face is 45 degrees. The real depth is about $\frac{3}{4}$ the altitude of the triangle.

The relative advantages of the various forms of threads; and the uses to which they should be put may be understood by considering the forces which act on the faces.

It has been proved that the greater the angle of the screw

thread the greater the friction between the bolt and nut, and the greater the force tending to burst the latter.

The friction of the square thread is less than that of the triangular thread because the angle between the sides is zero; there is moreover, no force tending to burst the nut. The triangular thread is, however, nearly twice as strong as the square thread. For the above reasons the square thread is better for transmitting motion and the triangular for fastening.

The trapezoidal thread should be used only when the pressure comes on the side perpendicular to the axis. In this case the thread has the same friction as the square thread and the same strength as the V thread. If the pressure is put on the inclined side the friction and bursting force are greater than is the case with the Sellars thread having angles of 60° between the sides.

Threads are formed for both *right handed* and *left handed* motion. Usually they are right handed. To determine the motion, hold the bolt or screw horizontal and turn it in the direction in which the hands of a watch revolve. If it advances into the nut or wood it is right handed. If when vertical the slope of the thread is from right to left it is right handed. For nuts reverse the above rule.

Multiple Threads. In tracing the thread about the screw, the next thread is reached in one revolution, if the screw is single threaded. In other words, the nut will advance a distance equal to the pitch for every revolution of the screw. If in tracing the thread through a turn one thread is missed, it is a double threaded screw; if two are missed it is a triple threaded screw, and so on.

Multiple threaded screws are used for transmitting motion, when it is desirable to have the nut advance a considerable distance for each revolution. This could also be accomplished by making the pitch large; but the multiple thread is better. The diameter at the root of the multiple thread is greater than that of a single thread and therefore stronger. The pitch is the distance the screw advances during one revolution, or it is the distance between two consecutive threads, if double threaded; or the distance between three threads, if triple.

Gas Pipe Threads. The Sellars thread is not suitable for the threads on gas pipe, for the calculated depth of thread would

be greater than the thickness of the pipe. For this work a special system has been adopted having smaller pitch and cutting less deeply into the metal.

Proportions of Bolts and Nuts. The diameter of the bolt determines the dimensions of the nut. These dimensions may vary to suit circumstances. Sometimes in cramped places the nut must be made thin, or there must be little metal around the screw threads, or it must be made of peculiar shape. In altering the shape or size of a nut, the designer considers the strain put on it. The standard form is shown in Fig. 11. The head of the bolt is square. Sometimes the neck (the portion next the head) is made square also, to prevent rotation of the bolt when the nut is being screwed up.

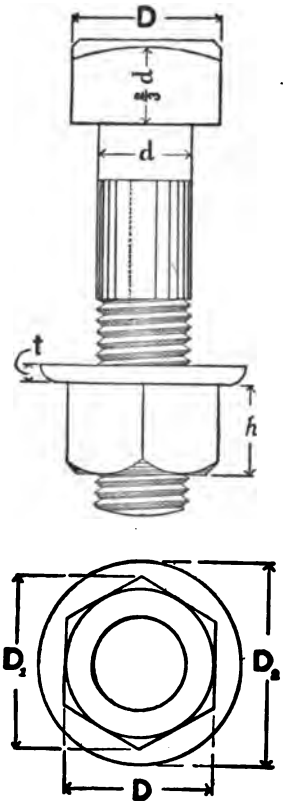


Fig. 11.

The nut is hexagonal and the washer circular. The washer is used with rough castings to give a smooth surface on which to turn the nut. The following are the formulas for dimensions corresponding to the figure, d being the diameter of the bolt.

For Rough Work.	For Finished Work.
$D = 1\frac{1}{2}\,d + \frac{1}{8}$ "	$D = 1\frac{1}{2}\,d + \frac{1}{16}$ "
$D_1 = 1.73d + .14$ " for hexagonal	$D_1 = 1.73\,d + .07$ " for hexagonal
$D_1 = 2.12\,d + .18$" for square	$D_1 = 2.12\,d + .09$ " for sq.
$D_2 = 1\frac{1}{8}\,D_1$	$D_2 = 1\frac{1}{8}\,D_1$
$h = d$	$h = d - \frac{1}{16}$ "
$t = .15\,d$	$t = .15\,d$

STRENGTH OF SCREW BOLTS.

Bolts are generally used when the straining force is in the direction of the axis of the bolt; that is, bolts are used for *tension stresses*. It is evident that the effective area is not the area of the cross-section of the bolt, but the area at the root of the thread.

Let P = the total load on the bolt.

d_1 = diameter at root of thread.

a = area of cross-section at root of thread.

S_w = safe working stress in pounds per square inch.

Then for tension

$$P = a S_w = \frac{\pi d_1^2 S_w}{4}, \text{ from which } a = \frac{P}{S_w}.$$

$$\text{Then } d_1 = 2\sqrt{\frac{P}{\pi S_w}}.$$

The values of a are found directly from the preceding table.

The value of P we can usually calculate from the machine. S_w varies with the material and the conditions of stress; if it is constant a good wrought iron bolt will stand 7,000 or 8,000 pounds per square inch. For variable stresses, S_w may be taken as about 5,000 or 6,000 pounds. Usually S_w is taken as 4,000 or 5,000 pounds. For bolts used in cylinder heads, S_w varies from 3,000 pounds per square inch for small to 6,000 pounds for large cylinders.

Suppose we wish to find the diameter of a bolt to sustain a steady stress of 15,000 pounds, allowing 8,000 pounds as the working stress.

$$a = \frac{P}{S_w} = \frac{15{,}000}{8{,}000} = 1.875 \text{ square inches.}$$

The number 1.875 lies between 1.7437 and 2.0485 of the table. The larger value should be chosen, the diameter of the bolt being $1\frac{1}{8}$ inches.

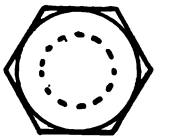
EXAMPLES FOR PRACTICE.

1. What is the safe working stress on a bolt 1 inch in diameter, if the value of P is 3,850 pounds?

Ans. 7,000 pounds (about).

2. Find the size of bolt used for varying stress. $P = 14{,}000$ pounds and $S_w = 4{,}000$ pounds. Ans. $2\frac{1}{2}$ inch bolt.

3. An engine cylinder head is bolted to the cylinder by 12 bolts. If the total steam pressure is 48,000 pounds, what is the diameter of the bolts? S_w being 4,250 pounds. Ans. $1\frac{3}{8}$ inches.



It would take considerable time to make the threads of all the screws and bolts of working drawings accurately. To save time a **conventional form**, shown in Fig. 12, has been adopted. The threads are represented by alternately light and heavy lines. The distance between these lines need not be equal to the pitch of the threads, because, the diameter being given, the number of threads per inch is found from the table. If the threads are not standard the number of threads per inch is noted on the drawing.

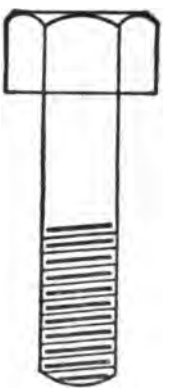


Fig. 12.

WRENCHES.

Forms of solid wrenches or spanners are shown in Fig. 13. The dimensions are given in decimals of the diameter of the bolt. They

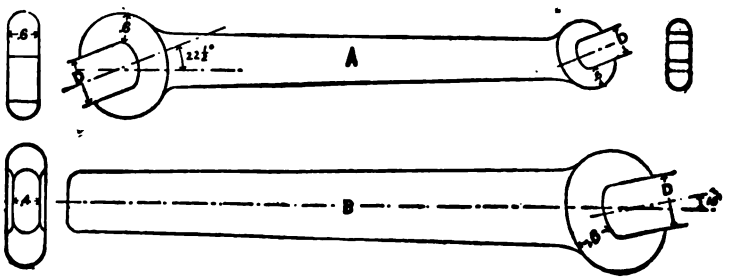


Fig. 13.

are made in many sizes, with ends shaped for hexagonal and square nuts. The unit for the proportions is D.

FORMS OF NUTS.

The most common form of nut is the hexagonal shown at A, Fig. 14; B shows the square nut. Usually both square and

hexagonal nuts are chamfered off at an angle of 30 to 45.° Sometimes they are finished with a spherical bevel, having a radius of about twice the diameter of the bolt. C shows a round nut having holes in the sides into which a bar is inserted for tightening

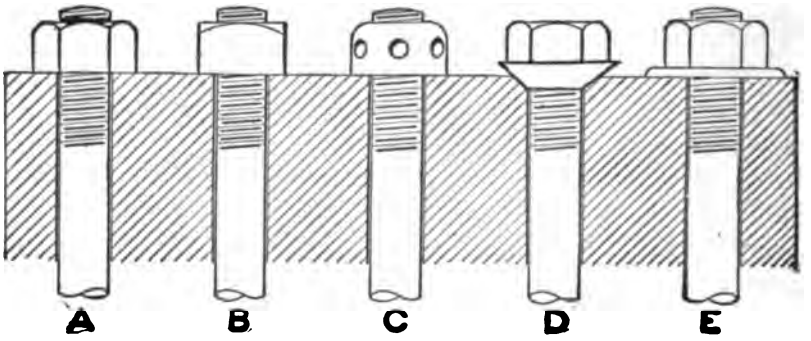


Fig. 14.

the nut. The nut shown at D is called a cap nut; it is used to prevent leakage past the screw thread. A thin copper washer is sometimes used with this form of nut. E represents a flange nut which is used when the hole in which the bolt is placed is considerably larger than the bolt. The flange covers the hole and gives greater bearing surface.

FORMS OF BOLT HEADS.

Fig. 15 shows several forms of bolt heads. The hemispherical or cup-shaped head, a common form, is shown at A. At B is

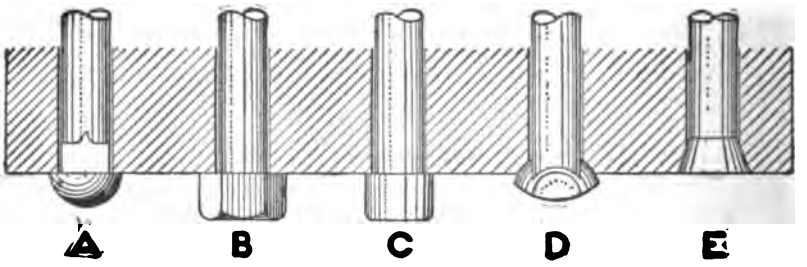


Fig. 15.

shown the hexagonal form. It is similar to the hexagonal nut and has about the same dimensions except the height which is usually less; it is from $\frac{2}{3}$ d to d. The cylindrical bolt head is

shown at C; rotation of the bolt is prevented by the square neck. D shows the spherical head; the bearing surface rests on a seat of the same shape. It is used when the bolt tends to lean toward one side. The head remains in contact with the seat for every position of the bolt. E shows a bolt with countersunk head; rotation is often prevented by a set-screw.

Fig. 16 is the **hook** bolt. It is used when a piece would be

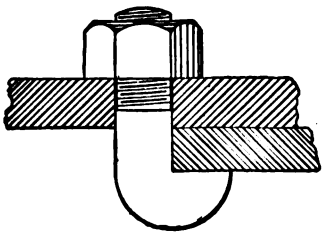


Fig. 16.

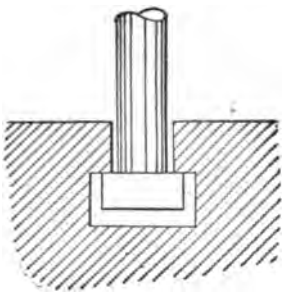


Fig. 17.

weakened or is too small to have a drilled hole. Fig. 17 is a **T-headed** bolt.

Set-screws are screws or bolts used to prevent by friction relative rotation between pieces. Set-screws are often used to prevent the hub of a pulley from turning on the shaft. They

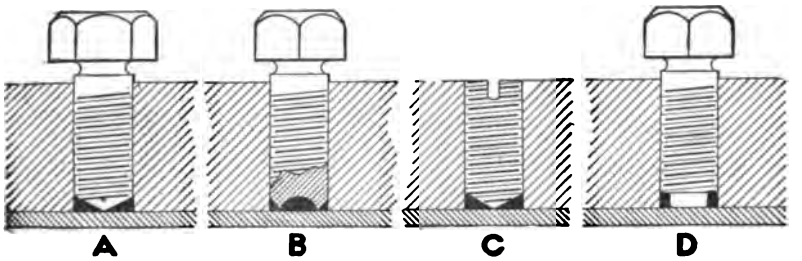


Fig. 18.

are screwed through the hub and prevent rotation by pressing against the shaft. At A, Fig. 18, is shown the cone point set-screw. The one shown at B is called the cupped set-screw; C, the headless cone point set-screw; and D, the round point set-screw.

A **stud bolt** is one that has threads cut on both ends. One

end is screwed into one of the pieces to be connected and remains in position when the nut is on or off. Fig. 19 shows a stud bolt. They are sometimes used for cylinder-heads and valve-chest covers. A stud with a **collar** is shown in Fig. 20. The collar may be square or round; if square it is a convenient place for a wrench. The collar forms a shoulder against which the stud may be screwed.

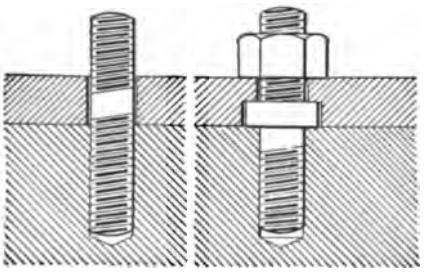


Fig. 19. Fig. 20.

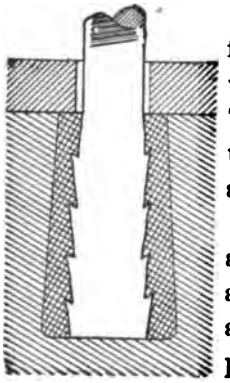


Fig. 21.

Fig. 21 shows an ordinary method of fastening a bolt to **stonework.** The head, which is long, is made jagged with a cold chisel. The hole is made larger at the bottom than at the top and after the head is placed, the space around it is filled with melted lead or sulphur.

Foundation bolts, which are used to fasten an engine-bed to its foundation, are often fixed as shown in Fig. 22. The head is formed by a cast iron washer and a cotter. This cotter passes through a slot and has gib ends to prevent slipping. The washer provides a large bearing surface. The bolt, washer, and cotter are placed in a recess in the wall and are accessible. The size of the washer is easily determined. The area of the washer multiplied by the compressive or crushing strength of the stone or brick should be equal to the pull on the bolt. The tensile strength of wrought iron is about 55,000 pounds per square inch and the compressive strength of

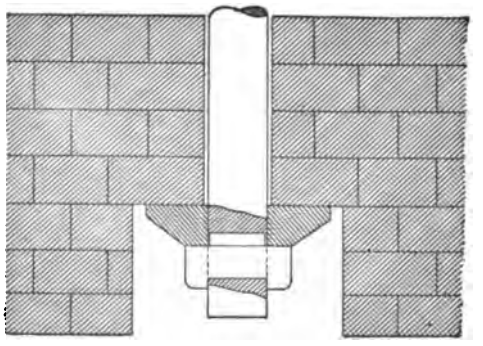
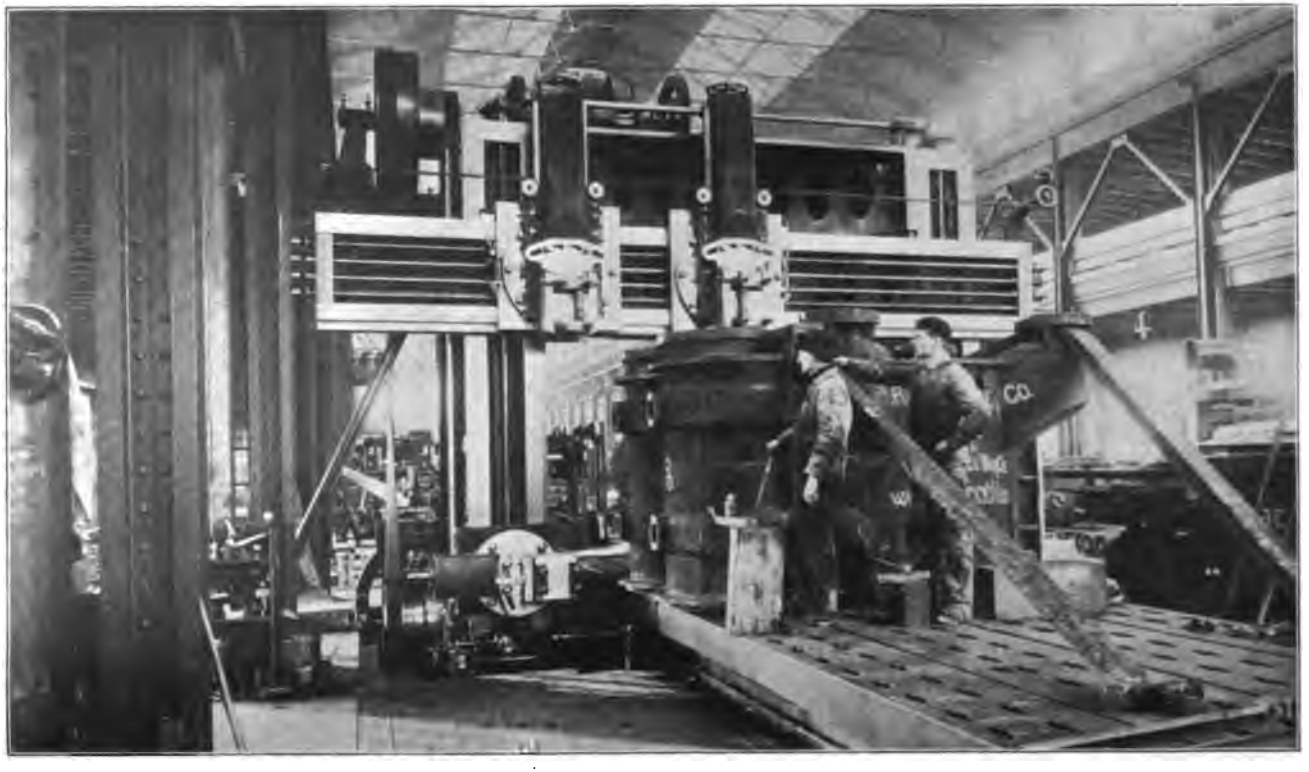


Fig. 22.



The 30-foot planer, planing a 19-ton casting in the Fore River Ship and Engine Company's machine shop.

brick is about 2,500 pounds per square inch, or about $\frac{1}{22}$ of that of the bolt. Therefore, the bearing surface of the washer should be about 22 times the area of the cross-section of the bolt.

Screws. The three most common forms of machine screws

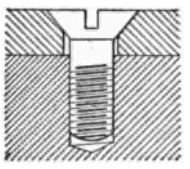


Fig. 23.

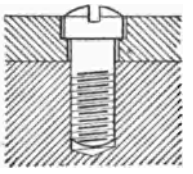


Fig. 24.

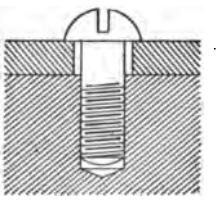


Fig. 25.

are shown in Figs. 23, 24, and 25. Fig. 23 is the **countersunk** head screw; Fig. 24, the **fillister head** and Fig. 25 the **button head.** The countersunk head is used when the head is not to project above the plate.

LOCKING ARRANGEMENTS FOR NUTS.

Nuts never fit the bolt accurately; some clearance is necessary to permit them to turn freely. If a nut is subject to frequent changes of load and vibration it gradually unscrews or slacks back. To prevent nuts from becoming loose various locking devices are used.

One of the most common is the double nut called a **locknut**

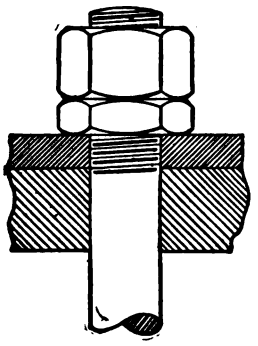


Fig. 26.

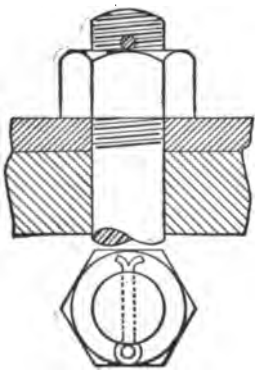


Fig. 27.

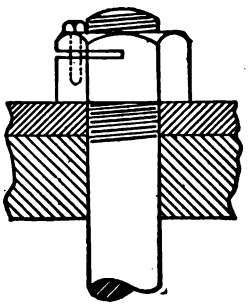


Fig. 28.

or jam nut shown in Fig. 26. There are two nuts, one about

twice as thick as the other. The outer nut should be the thicker because the load is thrown on it. However, the thin one is often placed on the outside because the wrench is often too thick to act on it when it is inside. When the nuts are screwed home they are locked together by being turned in opposite directions.

In Fig. 27 the nut is kept in place by a **split pin** or cotter. A hole is drilled through the bolt and the pin driven through and the ends turned over to prevent it from backing out. This is not a very good method because the nut must always be close to the pin.

Another method is shown in Fig. 28. The nut is sawed about half way through and the parts closed slightly by a set-screw, after the nut is screwed home. The nut then grips the thread tightly. For small nuts the set-screw is not used but the parts are closed, just before it is set, by a slight blow of a hammer.

At A Fig. 29 is shown a **Grover's spring washer.** The upper

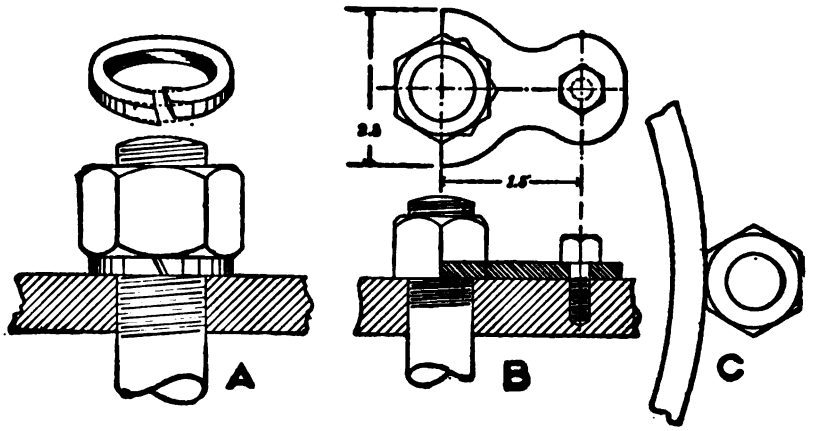


Fig. 29.

portion of the figure shows the washer when not held down by the nut. When the nut is screwed down tightly, the washer becomes nearly flat and its elasticity increases the friction between the threads of the bolt and nut.

In the device shown at B, Fig, 29, a **stop plate** is used. It is fixed by a set-screw to one of the pieces through which the bolt passes. It is shaped so that the nut may be locked at intervals of

$\frac{1}{12}$ of a revolution. The set screw may have a diameter equal to

$$\frac{d}{4} + \frac{1}{8}'',$$

d being the diameter of the bolt. The other dimensions are also in terms of the diameter. For bolts set in a circle, the bolts of a cylinder head for example, a **circular stop plate**, shown at C, Fig. 29, is used. It is placed inside the nuts and bears against one of the parallel sides.

There are numerous methods of locking by set-screws, their forms depending on the position of the pieces.

Usually bolts are used in tension, that is, when the straining force is parallel to the axis of the bolt. The joint pin is a

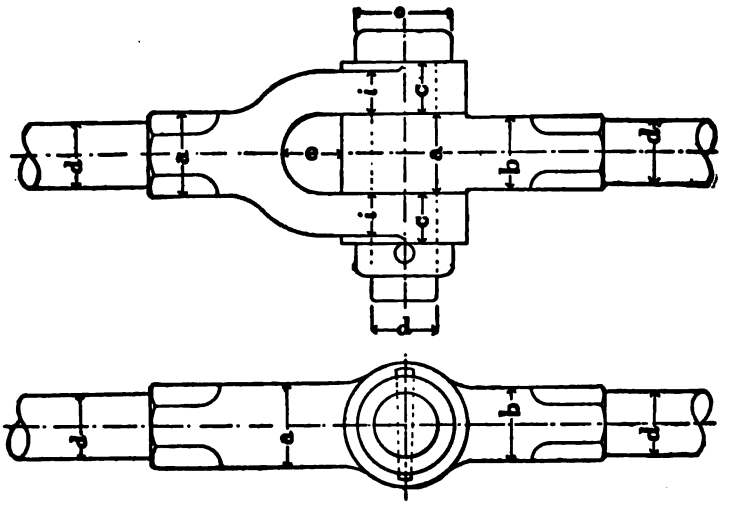


Fig. 30.

bolt placed so as to be in **shear**. Fig. 30 shows a **knuckle joint**. The joint pin is made the same size as the rods because of wear. If the pin were subjected to simple shear at the two sections it need be only about .7 the diameter of the rod, but it soon wears and is subjected to bending stresses also.

The other proportions are in terms of the diameter of the rods.

d = diameter of the rods.
$a = 1.2\ d$
$b = 1.1\ d$
$c = .75\ d$
$e = .8\ d$
$i = .6\ d$
$o = 1.5\ d$

Keys are small wedges, usually made of iron or steel, used to fix wheels, cranks and pulleys to shafts. It is the duty of keys to prevent the wheel or crank from rotating otherwise than with the shaft on which it is keyed. For example, a crank is keyed to the shaft in order that the shaft and crank will rotate together. Usually the friction of the key in the keyway will also prevent the wheel from sliding along the shaft.

Keys are usually plain rectangular pieces with their lateral sides parallel and a tapering thickness. In case the small end is

Fig. 31.

inaccessible, that is, the arrangement is such that it cannot be driven out, the key is made in the form shown in Fig. 31. A gib head is made at the large end which forms a shoulder to drive against.

Fig. 32 shows several forms of keys. The concave, or **saddle key**, is shown at A. The slot or keyway is cut in the wheel and the key hollowed to fit the shaft which is not cut at all. As the key holds by friction, it is suitable only for light work. The **flat**

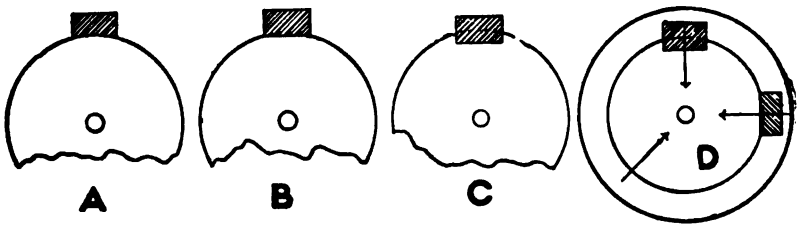


Fig. 32.

key is shown at B. A flat surface is planed on the shaft having a breadth equal to that of the key. This form is more secure than the saddle-key. The **sunk** key shown at C is more effective than either of the above forms because slipping is prevented unless the key shears. There are two forms of sunk keys, the rectangular and the square. The rectangular form is used for fastening

cranks, gear wheels, pulleys, etc. They are driven in tightly and fit both at the top and bottom as well as the sides. As they fit on all sides they should not be used in accurate work because they are likely to spring the work out of true. Square keys are used for accurate work; they fit accurately only on the sides. In case a pulley is accidentally bored a little too large for the shaft it is fastened securely by using both a flat key and a sunk key as shown at D. The two keys are placed at about right angles. The pulley and the shaft have a bearing at three points on the circumference of the shaft.

Sometimes large wheels are keyed to the shaft by two, three or four keys. When the shaft is square eight keys may be used.

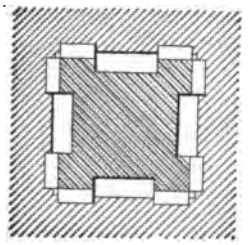


Fig. 33.

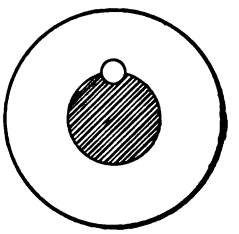


Fig. 34.

If they are arranged as shown in Fig. 33 no keyways need be cut on the shaft, and only the key seats are planed.

Pin-keys. A round taper pin is used in place of a key for small shafts; handles on valve stems for instance. The pin is sunk half in the shaft and half in the piece, as shown in Fig. 34.

Sometimes a secure connection, where the parts are not to be separated, is desired, as is the case with small cranks. The crank is bored slightly smaller than the crank shaft, then expanded by heat and shrunk on. It is further secured by a key or pin.

In order that keys may be easily driven in and removed they are tapered slightly. The taper is about 1 in 64 to 1 in 150. The more accurate the work the less the taper. This method of expressing taper means that the decrease in thickness is $\frac{1}{64}$ to $\frac{1}{150}$ of the length. Sometimes taper is expressed in inches to the foot as $\frac{1}{8}$ inch per foot.

Sliding Keys. Sliding or feather keys are used if the piece

is to be prevented from rotating but at the same time is to be allowed to slide along the shaft. The key may be fastened to the piece and free to slide in the keyway of the shaft, or it may be fast to the shaft and the wheel free to slide. Figs. 35 and 36 show the various methods of fixing the key. In Fig. 36 the key

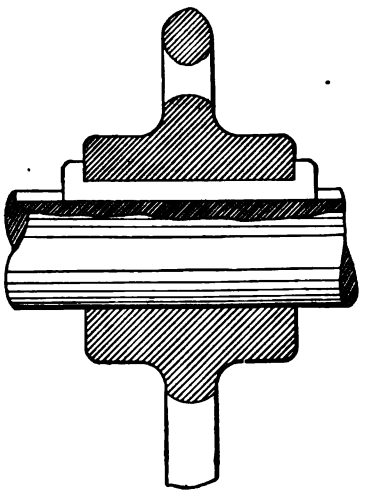


Fig. 35.

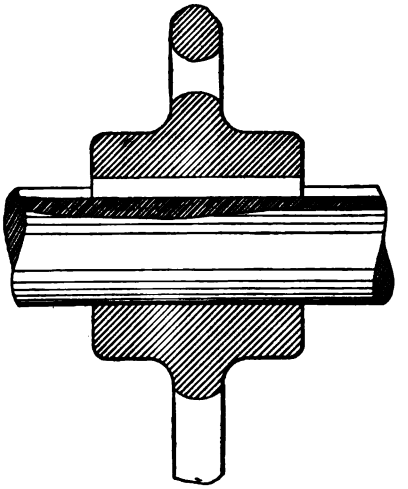


Fig. 36.

is dove-tailed in section and is fastened to the hub. The one shown in Fig. 36 is used when the hub comes against a collar or bearing, since it does not project from the hub. The feather key of Fig. 35 has gib heads.

Strength of Keys. Saddle keys are used most where the stress between the pieces is usually small. No exact rules can be given as they depend on friction to prevent rotation.

Sunk keys must resist both shear and compression. The twisting of the shaft tends to shear the key and crush it.

Let b = width of key in inches.
l = length of key in inches.
t = thickness of key in inches.
S_s = allowable shearing stress in pounds per square inch.
S_c = allowable crushing stress in pounds per square inch.
d = diameter of the shaft in inches.
r = radius of wheel or pulley in inches.
F = force in pounds acting at the rim.

The resistance to shearing is the shearing area multiplied by the shearing stress, blS_s. To find the diameter of the key, take moments about the center of the shaft and solve for bl

$$\tfrac{1}{2}\, d \times blS_s = Fr, \text{ from which } bl = \frac{2Fr}{dS_s}.$$

The sunk key usually has $\frac{1}{2}$ the thickness in the shaft and $\frac{1}{2}$ in the hub. In case the key is designed to be equally strong to resist shearing and crushing the shearing strength must be equal to the crushing strength.

The resistance to shearing is blS_s; and the resistance to crushing is equal to the product of the bearing surface and the stress, or $\frac{1}{2}\, tlS_c$.

If the crushing stress is assumed to be twice the shearing stress, $S_c = 2S_s$, then $b = t$ and the key is square in section. Usually the key is made wider than the thickness so that the shearing strength shall be greater than the resistance to compression, as there is little danger of crushing.

Let H. P. = the horse-power transmitted.

N = the number of revolutions per minute.

r = the radius of wheel in inches.

Then as the circumference of a circle is $2\pi r$, a point on the circumference will move $2\pi rN$ inches or $\frac{2\pi rN}{12}$ feet per minute.

The power transmitted (assumed to be constant) will be the force acting, F, multiplied by the distance $= \frac{2\pi rN}{12}$ or,

$$F \times \frac{2\pi rN}{12} = \text{foot-pounds.}$$

One H. P. = 33,000 foot-pounds per minute.

$$\text{Then H. P.} = \frac{2\pi rNF}{12} \div 33{,}000 = \frac{\pi rNF}{198{,}000}$$

$$\pi rNF = 198{,}000 \times \text{H. P.}$$

$$\text{or,} \quad Fr = \frac{198{,}000 \text{ H. P.}}{\pi N}$$

$$Fr = 63{,}025\ \frac{\text{H. P.}}{N}$$

For shearing of keys the usual factor of safety is about 10 which gives an allowable stress of about 5,000 pounds per square inch for wrought iron and about 7,000 pounds per square inch for steel. Inserting these values in the equation $bl = \frac{2Fr}{dS_s}$,

$$bl = \frac{Fr}{d \times 2{,}500}, \text{ for wrought iron}$$

$$= \frac{Fr}{d \times 3{,}500}, \text{ for steel.}$$

Then $$bl = \frac{25.2 \text{ H. P.}}{d \times N}, \text{ for wrought iron}$$

$$= \frac{18 \text{ H. P.}}{d \times N}, \text{ for steel.}$$

Suppose the pressure on a crank-pin is 15,000 pounds and the crank is 12 inches long. If the shaft is 6 inches in diameter and the length of key 6 inches, what should be the dimensions of the wrought iron key?

$$bl = \frac{Fr}{2{,}500d}$$

$$b = \frac{Fr}{2{,}500 \times l \times d} = \frac{15{,}000 \times 12}{2{,}500 \times 6 \times 6} = 2 \text{ inches}$$

Then the key should be 2 inches broad. Its thickness is usually some fraction of the breadth. We will make it $\frac{3}{4}$, or $1\frac{1}{2}$ inches. Keys vary in thickness from $\frac{1}{2}$ to $\frac{5}{6}$ of the breadth.

In designing machinery, the dimensions are usually determined by empirical formulas. For the ordinary sunk key $b = \frac{1}{4}d + \frac{1}{8}''$, and the mean thickness, $t = \frac{1}{2}b$, to $\frac{5}{6}b$.

If we use these formulas for finding the dimensions of the key in the above example, $b = \frac{1}{4}d + \frac{1}{8} = \frac{1}{4} \times 6 + \frac{1}{8} = 1\frac{5}{8}$ inches. $t = \frac{3}{4}b = \frac{39}{32} = 1\frac{7}{32}$ inches.

When pulleys are keyed to large shafts which transmit only a small amount of power, the dimensions obtained from the above formulas are larger than necessary. In such cases the following formulas are used.

$$d = \sqrt[3]{\frac{100 \text{ H. P.}}{N}} \text{ or, } \quad d = \sqrt[3]{\frac{Fr}{360}}.$$

These values of d are inserted in the above formulas.

A pulley transmits 3 horse-power. It is keyed to a shaft 6 inches in diameter, which makes 130 revolutions per minute; find the dimensions of the key.

$$d = \sqrt[3]{\frac{100\,\text{H. P.}}{\text{N}}} = \sqrt[3]{\frac{300}{130}} = 1.32 \text{ inches.}$$

$$b = \tfrac{1}{4}d + \tfrac{1}{8}'' = \frac{1.32}{4} + \tfrac{1}{8}'' = .455 \text{ or } \tfrac{1}{2} \text{ inch (about).}$$

$$t = \tfrac{1}{2}\,b = \tfrac{1}{4} \text{ inch.}$$

EXAMPLES FOR PRACTICE.

1. A shaft is 4 inches in diameter; what is the breadth of the sunk key? Ans. $1\frac{1}{8}$ inches.

2. Find the breadth of a steel key, when the length of the key is 4 inches, the horse-power transmitted is 100, the shaft is $4\frac{1}{2}$ inches in diameter and makes 100 revolutions. Ans. 1 inch.

COTTERS.

A **cotter** is an iron or steel bar driven through one or both of two pieces to be connected. It prevents their separation by its resistance to shearing at two transverse cross-sections. Cotters sometimes adjust the length of the pieces connected. They should be so designed as to decrease as little as possible the strength of the connected pieces.

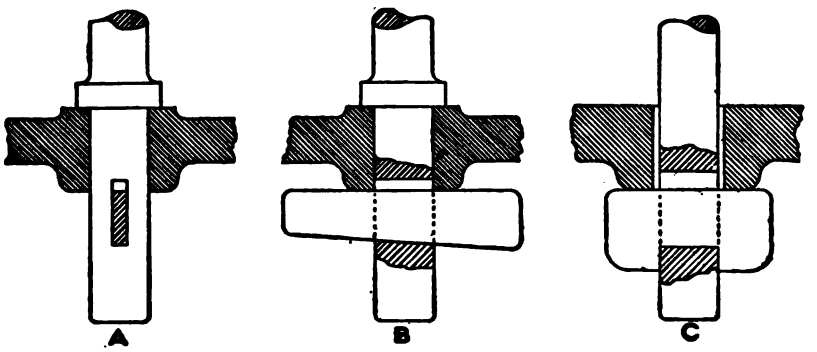


Fig. 37.

A and B of Fig. 37 show two views of a simple cotter. In this form the cotter passes through the rod. The cotter resists tension. The collar or enlargement prevents movement in the

other direction, and therefore resists thrust. An arrangement designed to resist tension only is shown at C, Fig. 37. The cotter has **gib ends** to prevent its moving out of place.

A construction to resist tension alone is shown at D and E of Fig. 38. This cotter is divided; one part with hooked ends is called

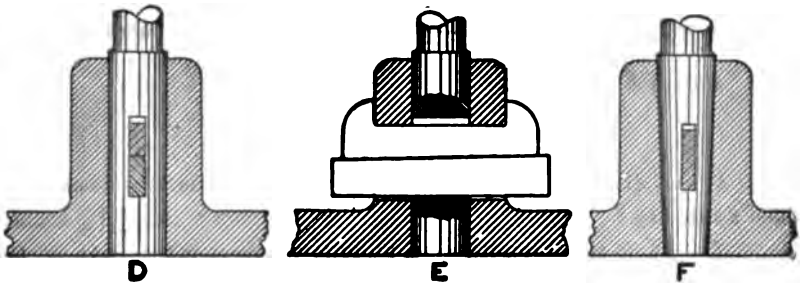


Fig. 38.

the gib and the other a plain cotter. Such a construction is often called the gib and cotter. At F of the same figure, the rod is tapered to provide for thrust.

Cotters like those shown at D and E in Fig. 38 are long and tapered and therefore may be used to adjust the length of the connected pieces. By driving the cotter in, the total length of the pieces is made less.

A cotter is often used to connect two straps, *a* and *b*, to the rod *l*, as shown in Fig. 39. If a plain cotter is used the excessive friction between the cotter and the straps, when the former is driven down, causes the lower strap to open as shown by the dotted lines. To prevent this a gib and cotter are used, or two gibs and a cotter. The gib prevents the strap from spreading. In Fig. 40 the side, *a b*, of the gib and the side, *c d*, of the cotter are parallel to each other and are at right angles to the straps. The parts of the gib and cotter that are in contact are tapered.

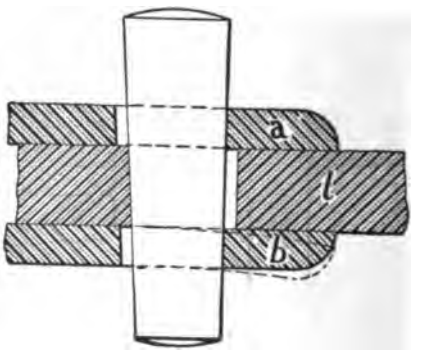


Fig. 39.

Taper of Cotters. A cotter that has a taper of more than 1 in 7 is likely to slack back. In general, the taper is from 1 in 24 to 1 in 48. If the cotter has a fastening device it may have a much greater taper, *i. e.*, 1 in 6 or 1 in 8. The taper of the cotter is found by dividing the increase in width by the length. Thus if a cotter is $2\frac{1}{4}$ inches in width at one end and $2\frac{3}{4}$ at the other the increase is $\frac{1}{2}$ inch and if the cotter is 10 inches long the taper is $\frac{1}{2}$ divided by $10 = \frac{1}{20}$ or 1 in 20.

Strength and Proportions of Cotters. In designing cotters a few facts must be remembered.

In the following demonstration the letters refer to Fig. 42.

The **cross-section,** $b\ t$, must be sufficient to stand the shearing stress.

The **thickness,** t, must be large enough to prevent crushing.

The **diameters** should be so designed that the rod will not be weakened by the cutting of the slot for the cotter.

Let F = the force in pounds on the rod,

S_t = allowable tensile stress of the rod in pounds per square inch,

S_c = allowable compressive stress of the cotter or rod in pounds per square inch,

S_s = allowable shearing stress of the cotter in pounds per square inch,

The net area of the rod is the area of a cross-section minus the area of the slot or,

$$\text{net area} = \frac{\pi\, d^2}{4} - d\, t\,.$$

The shearing area of the cotter is $2\ b\ t$. The area subject to crushing is $d\ t$. The area of the socket subject to tension is $\frac{\pi}{4}\ (D^2 - d^2) - (D - d)\ t$. The area of the rod itself is $\frac{\pi\, d_1{}^2}{4}$.

Then, as each of these values when multiplied by its

respective allowable stress must be equal to the force, F,

$$F = \left\{ \frac{\pi d^2}{4} - d\,t \right\} S_t, \qquad (a)$$

$$F = 2\,b\,t\,S_s, \qquad (b)$$

$$F = d\,t\,S_c, \qquad (c)$$

$$F = \left\{ \frac{\pi}{4}(D^2 - d^2) - (D - d)\,t \right\} S_t, \qquad (d)$$

$$F = \frac{\pi}{4} d_1^{\,2}\, S_t, \qquad (e)$$

If the cotter is subjected to a force in one direction only, the allowable stresses may be,

Wrought Iron	$S_t = 10{,}000$	$S_s = 8{,}000$	$S_c = 20{,}000$
Cast Iron	$S_t = 2{,}800$		$S_c = 5{,}600$
Steel	$S_t = 13{,}200$	$S_s = 10{,}600$	$S_c = 26{,}400$

In case the forces act alternately in opposite directions the stresses are found by dividing the above values by 2.

The above values show that,

$$\frac{S_s}{S_t} = \frac{4}{5} \text{ and } \frac{S_c}{S_t} = 2, \qquad \text{then} \qquad \frac{S_c}{S_s} = \frac{5}{2} = 2\tfrac{1}{2}.$$

Then combining equations (b) and (c) and letting $t = \frac{1}{4}\,d$ we have,

$$2\,b\,t\,S_s = d\,t\,S_c$$

$$\text{or} \quad b = \frac{d\,S_c}{2\,S_s} = \frac{5}{4}\,d = 1.25\,d.$$

Combining (a) and (c)

$$\frac{\pi d^2}{4} - d\,t = d\,t\,\frac{S_c}{S_t}$$

$$t = \frac{\pi d}{12} = \frac{1}{4}\,d \text{ (about)}.$$

Combining (a) and (d) and taking $t = \frac{\pi d}{12}$,

$$\left\{ \frac{\pi d^2}{4} - \frac{\pi d^2}{12} \right\} S_t = \left\{ \frac{\pi}{4}(D^2 - d^2) - (D - d)\,\frac{\pi d}{12} \right\} S_t$$

$$D = \frac{4}{3}\,d.$$

To have the same tensile strength D should equal $\frac{4}{3}\ d$ but to prevent crushing, the bearing surface of the socket must equal that of the rod, or

$$(D - d)\ t = d\ t,$$

and $$D = 2\ d$$

Combining (a) and (e) and, as before, taking $t = \frac{\pi\ d}{12}$,

$$\left\{ \frac{\pi\ d^2}{4} - \frac{\pi\ d^2}{12} \right\} S_t = \frac{\pi\ d_1{}^2}{4}\ S_t,$$

$$d_1 = .816\ d.$$

Since the bearing surface of the collar should be equal to that of the cotter in the rod,

$$\frac{\pi}{4}\ (d_2{}^2 - d^2) = d\ t, \text{ and since}$$

$$t = \frac{\pi\ d}{12},$$

$$d_2 = 1.15\ d.$$

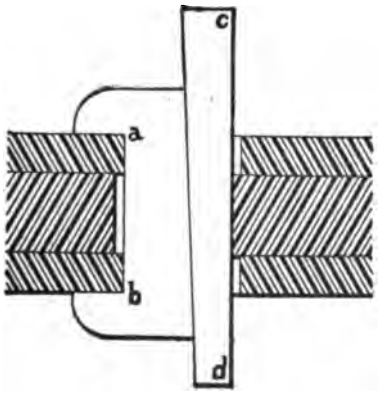


Fig. 40.

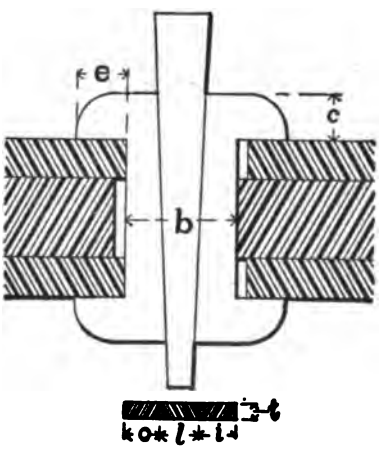


Fig. 41.

Tabulating the above results we find,

$$b = 1\tfrac{1}{4}\ d$$

$$t = \tfrac{1}{4}\ d$$

$$D = 2\ d$$

$$d_1 = .816\ d$$

$$d_2 = 1.15\ d$$

a is made equal to c and is from $\frac{3}{4}$ to $1\frac{1}{4}\ d$.

Fig. 41 shows a cotter with **double gib.** In this case $b\ t$ is usually made $\frac{5}{4}$ the sectional area of the strap and $t = \frac{1}{4}\ b$. b is made the same for all cases, whether a single cotter, gib and cotter or two gibs are used. The other dimensions are,

$$e = \frac{b}{8}$$

$$e = \frac{b}{4}$$

$$i = \tfrac{5}{16}\ b$$

$$o = \tfrac{5}{16}\ b$$

$$l = \tfrac{3}{8}\ b$$

If a steel cotter is used in a wrought iron rod, b may equal d.

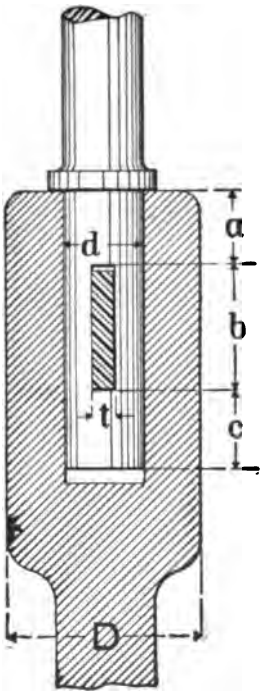


Fig. 42.

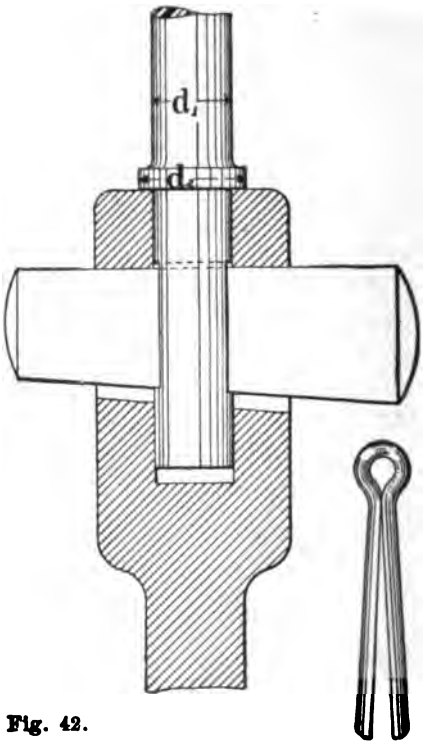


Fig. 43.

Fig. 43 shows a small **split pin.** Split pins are forms of cotters which are used to prevent two pieces from separating but

do not connect them firmly. Large pins are made solid with a slight taper.

Locking Arrangements for Cotters. Fig. 44 shows a method

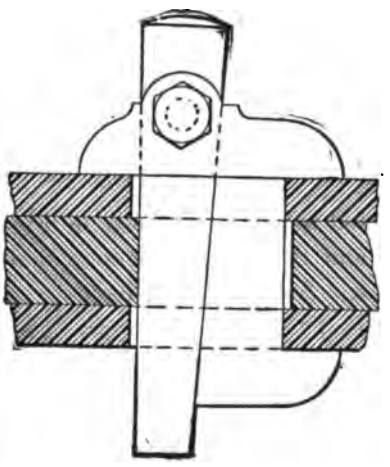


Fig. 44.

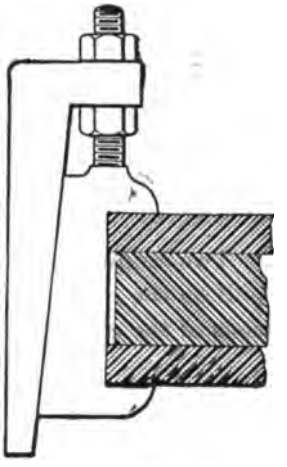


Fig. 45.

of securing the cotter by a set screw. The cotter passes through the head of the gib and is held by the screw. A simple way to secure the cotter is by prolonging the gib and having a screw

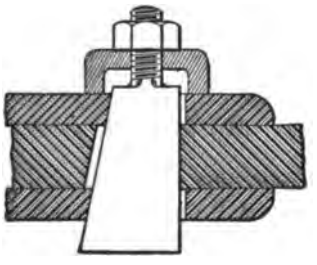


Fig. 46.

thread cut on it as shown in Fig. 45. Nuts on each side of the bent cotter keep it in place and adjust the length. In Fig. 46, the end of the cotter is a screw which passes through a recessed washer or extra seat. A nut above the washer holds it in place. In case a cotter has an excessive taper this method is sometimes used to prevent slacking back.

EXAMPLES FOR PRACTICE.

1. Find the dimensions of a rod and cotter of the form shown in Fig. 42. Assume $S_t = 6{,}500$ pounds. The load is 7,900 pounds.

Ans. $\begin{cases} d = 1\frac{1}{2} \text{ inches} \\ d_1 = 1\frac{1}{4} \text{ inches} \\ d_2 = 1\frac{3}{4} \text{ inches} \\ D = 3 \text{ inches} \\ t = \frac{3}{8} \text{ inch} \\ b = 1\frac{7}{8} \text{ inches} \\ a = 1\frac{1}{8} \text{ to } 1\frac{7}{8} \text{ inches} \end{cases}$

2. Two straps are connected to a rod as shown in Fig. 41. If the pull on the rod is 10,000 and $S_s = 5{,}500$ pounds, find the dimensions of the cotter and gibs. Assume $t = \frac{1}{2}$ inch.

Ans. $\begin{cases} t = \frac{1}{2} \text{ inch.} \\ \text{width of cotter} = 1\frac{1}{16} \text{ inches.} \\ \text{breadth of gibs} = \frac{9}{16} \text{ inches.} \\ b = 1\frac{13}{16} \text{ inches.} \end{cases}$

3. What are the dimensions of a wrought iron cotter used to fasten a wrought iron rod 2 inches in diameter?

Ans. $\begin{cases} t = \frac{1}{2} \text{ inch} \\ b = 2\frac{1}{2} \text{ inches} \end{cases}$

4. A cotter is $1\frac{1}{2}$ inches wide at the middle; it tapers on each side. If it is 18 inches long and tapers $\frac{1}{2}$ inch to the foot what is its width at each end?

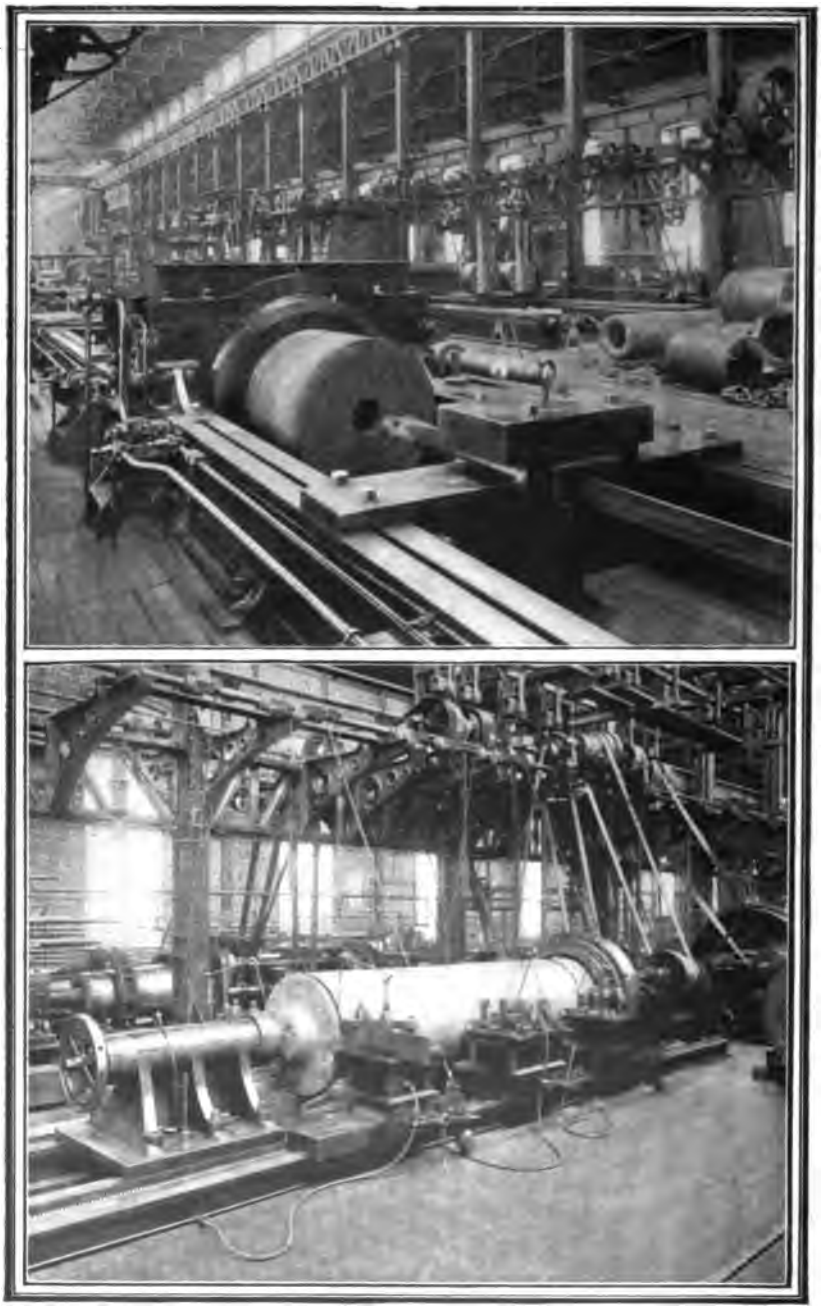
Ans. $1\frac{7}{8}$ inches and $1\frac{1}{8}$ inches.

JOURNALS.

Journals are the portions of shafts and axles which turn in bearings and are supported by the frame of the machine. They are usually cylindrical but may be conical or spherical in form.

If the journal is at or near the end of the shaft it is called an **end** journal; if situated between two end journals it is called a **neck** journal.

The most common form for the bearing portion of a journal is a true cylinder as shown in Fig. 47. Collars or shoulders at the ends bear against the ends of the brasses, in which the journal revolves, and limit the play lengthwise. In case a light end play



CUTTING AND BORING A FLUID-COMPRESSED INGOT
BETHLEHEM STEEL WORKS N. Y.

(By courtesy of the *Engineering Magazine*, New York and London)

is desired, as in the car journal for instance, the bearing is made slightly shorter than the journal, thus permitting a little longitudinal

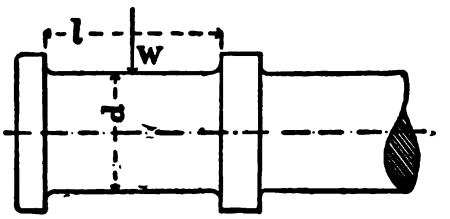


Fig. 47.

motion and causing uniform wear of the brasses. When longitudinal motion would interfere with the movements of other portions of the machine, it is made as small as possible.

The methods for calculating the requisite size of a journal depend upon the velocity of the shaft and the constancy with which it is run. In case it runs occasionally or at slow speed it is designed for strength; but if it runs constantly at high speed, durability and freedom from heating are as important elements as strength.

Journals may be subjected to straining forces in the plane of the axis (causing bending and shearing stresses) and also to combined torsional and bending stresses.

When designing for strength the journal is considered as a cantilever beam uniformly loaded.

If l = the length of the journal in inches,

d = the diameter of journal in inches,

S = the safe working stress,

I = the moment of inertia,

c = the distance of the fibre most remote from the neutral axis,

W = the total load in pounds,

w = load per square inch in pounds;

then from "Mechanics,"

$$W = 2\,\frac{S}{l} \times \frac{I}{c},$$

and since for a circular section,

$$I = \frac{\pi\, d^4}{64} \text{ and } c = \frac{d}{2}$$

$$\text{then,} \quad W = 2\,\frac{S}{l} \times \frac{2\,\pi\,d^4}{64\,d} = \frac{\pi\,d^3\,S}{16\,l}$$

$$\text{and,} \quad 16\,W\,l = \pi\,d^3\,S$$

$$\text{from which } d = \sqrt{\frac{16\,W}{\pi\,S} \times \frac{l}{d}}.$$

The equation is used in the above form because the ratio of length to diameter $\frac{l}{d}$ is usually assumed, *i. e.*, fixed before the calculation is made. If this ratio were not assumed there would be two unknown quantities in the equation.

For journals which work intermittently the ratio l to d is usually 1, or $\frac{l}{d} = 1$. Where the speed is less than 150 revolutions per minute $\frac{l}{d}$ varies from 1.5 to 1.75. The greater the speed the greater the proportion of length to diameter; this causes a reduction of pressure per unit of bearing surface as the speed increases.

For example. Find the length and diameter of a steel journal, having a load of 2,000 pounds. Assume $\frac{l}{d}$ to be 1.5 and the safe working stress, S, as 9,000 pounds.

$$d = \sqrt{\frac{16\,W}{\pi\,S} \times \frac{l}{d}} = \sqrt{\frac{16 \times 2{,}000}{3.1416 \times 9{,}000} \times \frac{3}{2}} = 1.3 \text{ inches.}$$

In this case the journal would be $1\frac{5}{16}$ inches in diameter and $1\frac{5}{16} \times 1.5 = 2$ inches long.

The safe working stress varies with the conditions and the material. Average values of S are as follows:

Material.	Constant Load.	Variable Load.
Steel	12,000 to 13,000	9,000 to 12,000
Wrought Iron	7,000 to 9,000	6,000 to 7,000
Cast Iron	3,500 to 4,500	3,000 to 4,000

When designing to provide against heating, experience determines the allowable pressure per square inch on the area of pro-

jection of the journal. This is called the projected area and is equal to the length of the journal multiplied by the diameter, or,

$$\text{area} = l \times d.$$

The load per square inch evidently is the total load divided by this area, or,

$$w = \frac{W}{l \times d}.$$

If the pressure per square inch w, is too large the lubricant is squeezed out and the journal is likely to heat and increase in size. The better the lubricant, the larger w may be. For high speeds the pressure may be greater than for low speeds. In the case of journals, as for example, crank pins, where the pressure alternates in direction the limit of pressure may be about twice as great as where the load is in a fixed direction. This is because of the better lubrication in case of alternating direction.

Since $W = w\ l\ d$, and $W = \frac{\pi\ d^3\ S}{16\ l}$;

then $w\ l\ d = \frac{\pi\ d^3\ S}{16\ l}$, or $\pi\ d^2\ S = 16\ w\ l^2$,

and $\frac{l^2}{d^2} = \frac{\pi\ S}{16\ w}$, or $\frac{l}{d} = \sqrt{\frac{\pi\ S}{16\ w}}$

Substituting this value of $\frac{l}{d}$ in the equation $d = \sqrt{\frac{16 W}{\pi\ S} \times \frac{l}{d}}$,

the result is, $d = 2\sqrt{\frac{W}{\sqrt{\pi\ S\ w}}}$,

and since $w\ l\ d = W$,

$$l = \frac{W}{w\ d}.$$

Therefore in calculating the diameter when the pressure w is assumed we use the formula $d = 2\sqrt{\frac{W}{\sqrt{\pi\ S\ w}}}$, and $l = \frac{W}{w\ d}$ for the length.

Find the length and diameter of a steel journal when the

load is 11,000 pounds. The safe working stress is 10,000 pounds and the allowable pressure is 900 pounds.

From the formula we obtain;

$$d = 2\sqrt{\frac{W}{\sqrt{\pi}\, S\, w}} = 2\sqrt{\frac{11{,}000}{\sqrt{3.1416 \times 10{,}000 \times 900}}}.$$

The square root of $\pi = 1.77245$, then $d = 2.87 +$ inches, and the shaft would be 3 inches in diameter. We also find

$$l = \frac{W}{w\, d} = \frac{11{,}000}{900 \times 3} = 4.07 = 4\tfrac{1}{16} \text{ inches long.}$$

The following table of limits of pressure per square inch of projected area for different conditions is taken from Unwin's Machine Design.

PRESSURE ON BEARINGS AND SLIDES.

Pressure Calculated in lbs. per sq. in. of Bearing Surface.	Intensity of pressure, lbs. per sq. in.
Bearings on which the load is intermittent and the speed slow, such as crank pins of shearing machines	3000
Cross-head neck journals	1200
Crank pins of large slow engines	800 to 900
Crank pins of marine engines, usually	400 to 500
Main crank shaft bearings Marine engines (slow)	600
Main crank shaft bearings Marine engines (fast)	400
Railway journals	300
Fly wheel shaft journals	150 to 250
Small engine crank pins	150 to 200
Slipper slide blocks, Marine engines	100
Stationary engine slide blocks	25 to 125
Stationary engine slide blocks, usually	30 to 60
Propeller thrust bearings	50 to 70
Shafts in cast-iron steps (Sellers)	15

NECK JOURNALS.

A neck journal is considered as a simple beam supported at both ends and uniformly loaded. The cross head pin, or wrist pin is an example of this kind of journal.

From "Mechanics," the equation for the above beam is,

$$W = 8 \frac{S}{l} \times \frac{I}{c}$$

$$\text{Since } I = \frac{\pi d^4}{64}, \text{ and } c = \frac{d}{2},$$

$$W = 8 \frac{S}{l} \times \frac{\pi d^3}{32},$$

$$\text{from which } d = 2\sqrt{\frac{W}{\pi S} \times \frac{l}{d}}.$$

This formula is used to calculate the value of d when the ratio $\frac{l}{d}$ is assumed.

Example. Find the diameter and length of a wrought iron neck journal when the load is 4,400 pounds, S assumed to be 8,000 pounds and $\frac{l}{d} = 2$.

$$\text{Solution,} \quad d = 2\sqrt{\frac{W}{\pi S} \times \frac{l}{d}} = 2\sqrt{\frac{4{,}400}{3.1416 \times 8{,}000} \times 2.}$$

$$= 2 \times .59 = 1.18 \text{ inches.}$$

A $1\frac{3}{16}$ inch shaft would probably be chosen.

Then, also, $l = 2\,d$, $2 \times 1\frac{3}{16} = 2\frac{3}{8}$ inches.

As in end journals $W = w\,d\,l$, and since $W = \frac{8\,S\,\pi\,d^3}{32\,l}$ equating the two values for W, we get,

$$w\,l\,d = \frac{8\,S\,\pi\,d^3}{32\,l},$$

$$\text{or,} \quad 4\,w\,l^2 = \pi\,d^2\,S.$$

$$\text{Then } \frac{l^2}{d^2} = \frac{\pi S}{4\,w}, \qquad \text{and } \frac{l}{d} = \sqrt{\frac{\pi S}{4\,w}}$$

Substituting this value of $\frac{l}{d}$ in the equation $d = 2\sqrt{\frac{W}{\pi S} \times \frac{l}{d}}$

we get, $d = 2\sqrt{\frac{W}{\pi S} \times \sqrt{\frac{\pi S}{4 w}}}$

and reducing, $d = 2\sqrt{\frac{W}{\sqrt{4 \pi S w}}}$,

as before, $l = \frac{W}{w d}$.

The same values for S and w, given for end journals may be used for neck journals.

Suppose we wish to find the length and diameter of a neck journal having a load of 8,000 pounds. Let us assume S to be 8,500 pounds and the bearing pressure 500 pounds; then,

$$d = 2\sqrt{\frac{W}{\sqrt{4 \pi S w}}} = 2\sqrt{\frac{8{,}000}{\sqrt{4 \times 3.1416 \times 8{,}500 \times 500}}}$$

$= 2\sqrt{1.09} = 2.08$ inches.

A $2\frac{1}{8}$ inch shaft would be used, and we also find

$$l = \frac{W}{w d} = \frac{8{,}000}{500 \times 2\frac{1}{8}} = 7\frac{1}{2} \text{ inches (about.)}$$

DIMENSIONS OF JOURNALS.

We have already seen how the diameter and length of journals are determined. Knowing the diameter, the height of the collar may be found from the formula $h = .1\ d + \frac{1}{8}''$ and the breadth equals $1\frac{1}{2}$ times the height.

Usually the journal is turned with a fillet in the corner, since the shaft is less likely to crack than if it has a square corner. The radius of the circular fillet is about one half the height.

FRICTION OF JOURNALS.

In every case, friction generates heat which causes the temperature of the journal to rise. This rise of temperature increases the size of the journal and causes unnecessary work.

In order to diminish friction, the journal is lubricated, that is, it is supplied with some lubricant (oil or fatty matter) which forms a thin film between the journal and the bearing. The lubricant diminishes friction and greatly reduces wear.

The surface velocity in feet per minute of a journal is evidently the circumference of the circle whose diameter is d, multiplied by the number of revolutions n, or expressed algebraically,

$$\text{Surface velocity} = \frac{\pi\, d\, n}{12} \text{ feet per minute.}$$

If f = the coefficient of friction, and W the load, the work expended in friction is expressed by the formula:

$$\text{Work} = f\ \text{W} \times \frac{\pi\, d\, n}{12} \text{ foot ponnds per minute.}$$

As we have seen from the table on page 54, the pressure per square inch on the bearing surface cannot exceed certain limits. These limits varying with the kind of work done and the speed. For the load W a certain bearing surface $d\,l$ is necessary.

The value of the coefficient f varies greatly; it is dependent upon speed, pressure, kind of metals, and kind of lubricant. Under some conditions it may be as low as .001 and under others .2.

From the above equation it is evident that the work used in friction is directly proportional to the diameter of the shaft. This fact shows that under a constant load it is well to obtain a large projected area, $d\,l$, by increasing l and making d as small as is consistent with strength.

Suppose we have two journals, $1\frac{1}{2}$ inches in diameter and $4\frac{1}{2}$ inches long; the other 3 inches in diameter and 2.25 inches long. They have the same projected area, *i.e.*, 6.75 square inches; but the work required to overcome friction of the first is only one-half as much as for the second.

Although the first, *i. e.*, the $1\frac{1}{2}$ inch journal is preferable for a steady load, it is not as strong as the three inch journal, and the latter should be used for high speeds and when the journal is subject to shocks.

PIVOT AND COLLAR JOURNALS.

In journals already considered the direction of pressure is perpendicular to the axis of the shaft; in the pivot journal shown in Fig. 48 the direction of pressure is parallel to the axis. With end journals the bearing surface is the cylindrical

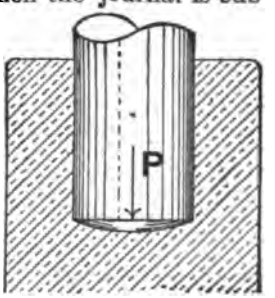


Fig. 48.

surface; but in pivot journals it is circular, and is the area of the end of the pivot. To find the diameter of such a journal knowing the load, assume a value for the pressure per square inch of bearing surface and solve for the area. This may be expressed by the formula,

$$W = \frac{\pi d^2}{4} \times w$$

Suppose the total load is 32,000 pounds and the allowable pressure on the bearing surface is 650 pounds per square inch. What is the diameter?

$$W = \frac{\pi d^2}{4} \times w, \text{ and } 32{,}000 = \frac{\pi d^2}{4} \times 650.$$

$$\text{Then } \frac{\pi d^2}{4} = \text{the area} = 49.23 \text{ square inches.}$$

and $d = 7.9$ inches.

An 8 inch shaft would be used.

In cases where the speed is not very high we may use the following as maximum values of w:

Wrought iron on gun metal	700 pounds
Cast iron on gun metal	470 pounds
Wrought iron or steel on lignum vitae	1,400 pounds

For high speeds with iron or steel on lignum vitae bearing, when moistened with water, the following formula may be used,

$$d = .035 \sqrt{W.}$$

The direction of load for a collar bearing is parallel to the axis but the bearing area is formed by a collar or collars on the shaft. Fig. 49 shows a journal with one collar and Fig. 50 one

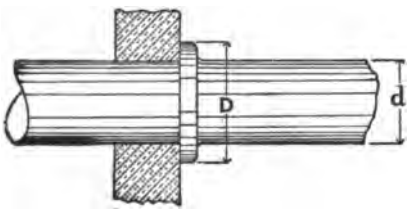


Fig. 49.

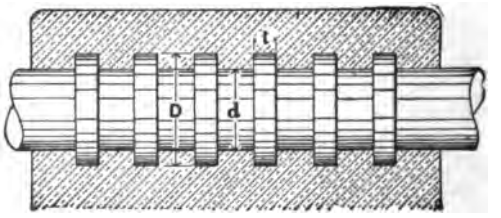


Fig. 50.

with several collars. To find the diameter of collars for propeller shafts we proceed as follows:

Let d = diameter of shaft,
D = diameter of collar,
N = number of collars,
w = pressure per square inch of projected area,
W = total load.

The total load is expressed by the equation,

$$W = \tfrac{1}{4}\pi\,(D^2 - d^2)\,N\,w.$$

Usually w is taken as 50 or 60 pounds for propeller shafts. Using $w = 60$ the formula becomes,

$$W = \pi\,(D^2 - d^2)\,15\,N,$$

$$\text{or} \qquad (D^2 - d^2) = \frac{W}{15\,\pi\,N}.$$

$$\text{Hence } D = \sqrt{d^2 + \frac{W}{15\,\pi\,N}}.$$

The number of collars is determined by the designer. By increasing the number, the diameter, wear and friction are decreased; but if too many collars are used all the thrust may be brought on a few of them. The distance between collars is sometimes equal to the thickness. But if the encircling rings are lined with white metal or are made hollow the distance may be made about twice the thickness.

EXAMPLES FOR PRACTICE.

1. What are the proportions of a wrought iron end journal, when $\frac{l}{d} = 1.2$, the load 1,800 pounds, and safe stress 8,500 pounds? Ans. $1\frac{3}{16}'' \times 1\frac{7}{16}''$.

2. Find the length and diameter of an end journal when S = 9,000 pounds, W = 8,000 pounds and w = 800 pounds.
Ans. $3\frac{13}{16}'' \times 2\frac{5}{8}''$

3. Find the proportions of a neck journal, if the load is 7,000 pounds, the bearing pressure 400 pounds and safe stress 9,000 pounds. Ans. $2\frac{1}{16}'' \times 8\frac{1}{2}''$.

4. Find the proportions of a neck journal if the load is 6,000 pounds, the safe stress 8,500 pounds and $\frac{l}{d} = 1.75$.
Ans. $1\frac{1}{4}'' \times 2\frac{3}{16}''$.

5. Find the power used in friction when the load is 6,000 pounds, the diameter of the shaft 2 inches, and making 150 revolutions per minute. The coefficient being .002.

Ans. 942.48 foot-pounds per minute.

6. Find the diameter of a wrought iron pivot running at 90 revolutions per minute, with a load of 850 pounds, having gun metal bearings. Ans. $1\frac{1}{4}$ inch shaft.

7. Find the diameter of the collars on an 8 inch shaft, the end thrusts being 18,000 pounds. There are four collars.

Ans. 12.6 inches. Use $12\frac{5}{8}$ inches.

SHAFTS.

Shafts are parts of machines which support rotating pieces. They are usually circular in section.

Shafts may be divided into three classes. The classification being determined by the kind of stresses to which they are subjected.

1. Shafts subjected chiefly to torsion or twisting. For example, shafting used to transmit power, called line shafting.

2. Shafts subjected chiefly to bending; shafts of gearing for example.

3. Shafts subjected to both torsion and bending, as engine shafts.

Shafts of the first class, those subjected to torsion, must be designed for both strength and stiffness. If a shaft is of large diameter or if it is short it may be designed for strength only; but for a long shaft of small diameter, sufficient strength may not insure sufficient stiffness. Line shafting is the name given to the long continuous lines of shafting used in mills, factories, etc. Numerous pulleys are keyed to the shafts from which power is taken by belts and gears. These shafts are strained by twisting stresses and also by a slight bending action due to the weight and the downward pull of the gearing.

Calculations for size are made by investigating the twisting moment which is equal to the force or pull multiplied by the radius.

From the study of "Mechanics" we know that the diameter of a round shaft may be found from the formula:

$$d = 68.5 \sqrt[3]{\frac{H}{n S_s}},$$

in which d = the diameter in inches,

H = the horse-power transmitted,

n = the number of revolutions per minute,

and S_s, the allowable shearing stress for the material, that is, the shearing strength per square inch divided by the factor of safety. The value to be used for S_s depends upon the nature of the stress as well as upon the material. Thus for a shaft subjected to a perfectly uniform twisting moment, S_s may be made as large as $\frac{1}{6}$ of the shearing strength; but for shafts which must transmit varying torsional stresses which may perhaps vary in direction, S_s must be made small.

A formula often used for ordinary wrought iron or steel mill shafting is:

$$d = \sqrt[3]{\frac{H}{n \times .01153}}.$$

This is derived from the above by giving to S_s a value of 8,700, which is low enough to allow for slight bending stresses of which the formula takes no account.

Hollow Shafts. When a shaft is subjected to torsion, the intensity of stress on any fibre is *directly proportional to the distance from the fibre to the centre of the shaft.* Consequently the stress is very much greater on the outer fibres than on the inner ones, and the outer ones are broken before the inner ones can exert their full strength and aid the outer ones in resisting the torsion. If the shaft is made hollow, without changing its weight, there will be the same number of fibres in its cross-section, and consequently the same shearing strength; but the distance from each fibre to the centre will be nearer to the average distance of all the fibres, and thus each fibre will bear more nearly its proper share of the load, so that the shaft as a whole will resist a greater torsion before any fibre breaks. Thus by making a shaft hollow, the same strength may be secured with a saving in weight.

Hollow shafts forged on a mandrel can be toughened through-

out by hammering, and then tempered throughout if desired, whereas a solid shaft subjected to the same processes would be affected but a small distance below the surface.

Let D = the outside diameter of the hollow shaft.

d = the diameter of a solid shaft having the same strength as the hollow shaft.

D_1 = the inside diameter of the hollow shaft.

From a mathematical consideration of moments and stresses the following formula is deduced:

$$D = d\sqrt[3]{\frac{1}{1 - \left\{\frac{D_1}{D}\right\}^4}}$$

Suppose we wish to find the diameter of a hollow shaft which shall have the same strength as a solid shaft of nine inches diameter, the internal diameter to be $\frac{1}{2}$ the external.

$$D = d\sqrt[3]{\frac{1}{1 - \left\{\frac{D_1}{D}\right\}^4}}$$

$$= 9\sqrt[3]{\frac{1}{1 - \left\{\frac{1}{2}\right\}^4}} = 9 \times 1.022 = 9.198 \text{ inches.}$$

The shaft may be made $9\frac{3}{16}$ inches in outside diameter and $4\frac{19}{32}$ in inside diameter.

The **distance between bearings** depends upon the number of pulleys on the shaft. If the shaft has several pulleys keyed to it the bearings must be nearer than if there are only a few pulleys. Large pulleys and those giving out a large amount of power should be placed near the bearings. The distance should be such that the deflection can not be more than $\frac{1}{100}$ of an inch per foot.

The following table gives maximum distance between bearings for shafts of various sizes.

Diameter of Shaft in inches.	Distance between Bearings in Feet.	
	Wrought Iron Shafts.	Steel Shafts.
2	15.46	15.89
3	17.70	18.19
4	19.48	20.02
5	20.99	21.57
6	22.30	22.92
7	23.48	24.13
8	24.55	25.23
9	25.53	26.24

SHAFT COUPLINGS.

Since it would be inconvenient to manufacture shafts long enough for a large factory, some means must be provided to join short lengths together.

Shafting is usually made in lengths of 20 to 30 feet, and joined by shaft couplings. These couplings should be placed near bearings and on the side farthest from the power. If this is done the running part is supported even if a length is disconnected.

Couplings are made in various shapes according to the positions of the shafts to be joined. They may be designed for shafts having a common axis of rotation, that is, in line; for parallel shafts; or for shafts whose axes intersect.

Shaft couplings are usually divided into three classes.

1. **Mixed or permanent** couplings, which can be disconnected only by slacking keys or by removing nuts.

2. **Loose or disengaging** couplings, which are provided with arrangements for throwing a part of the shafting out of gear with slight effort.

3. **Friction Couplings** which are loose and so arranged that they put the shafting into gear gradually and slip if the resistance is great.

The simplest form of shaft coupling is the box or muff shown in Fig. 51. A short iron cylinder is fitted over the ends of the shafts. Relative rotation is prevented by a wrought iron key which is usually half in the shaft and half in the coupling. The

shafts may be enlarged at the ends so that there will be no decrease in strength because of the key way. Couplings are

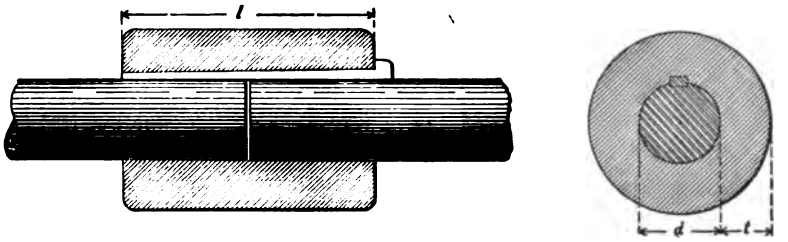


Fig. 51.

designed from empirical formulas and not from calculations for strength. In Fig. 51 d = the diameter of the shaft, and is taken as the unit.

The dimensions may be calculated from the formulas,

$$l = 2\tfrac{1}{2}\, d + 2''$$
$$\text{and } t = .45\, d + \tfrac{1}{2}''.$$

For the key,

$$b = \tfrac{1}{4}\, d$$
$$t = \tfrac{1}{6}\, d.$$

For example. Find the dimensions for a box coupling for a shaft 3 inches in diameter.

$$l = (2\tfrac{1}{2} \times 3) + 2'' = 9\tfrac{1}{2} \text{ inches}$$
$$t = .45\, d + \tfrac{1}{2}'' = 1.85 \text{ or } 1\tfrac{7}{8} \text{ inches.}$$

The key would be,

$$b = \tfrac{3}{4} \text{ inch}$$
$$t = \tfrac{1}{2} \text{ inch.}$$

The coupling shown in Fig. 52 is called the **clamp coupling.** It is made of cast iron, is easily removed, has no projecting parts

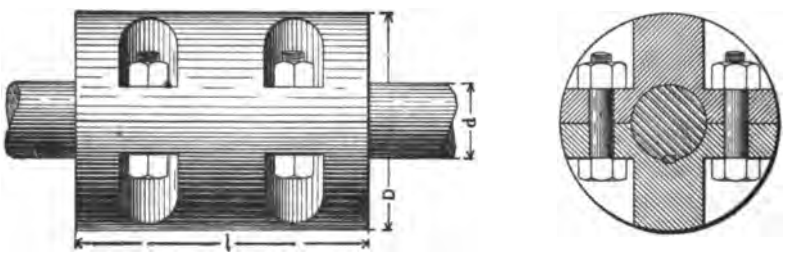


Fig. 52.

and on account of its cylindrical shape can be used as a pulley. The faces of the joint are first planed then the holes are drilled.

It is bored out after the two halves are bolted together with paper between them. When the paper is removed a slight space is left between the halves so that the coupling grips the shaft when the parts are bolted together. The key is straight and fits only at the sides so that it will not exert bursting pressure on the coupling. The following formulas may be used in finding the proportions; d being the unit.

$$D = 2\tfrac{1}{2}\, d + \tfrac{1}{2}''$$
$$l = 3 \text{ to } 4\ d.$$

The bolts may be $\frac{5}{8}$ inch in diameter for shafts under $2\frac{1}{2}$ inches in diameter and $\frac{3}{4}$ or $\frac{7}{8}$ inch for larger shafts. Usually four bolts are used for small and six bolts for large shafts.

Find the dimensions for a clamp coupling for a $3\frac{1}{2}$ inch shaft.

$$D = 2\tfrac{1}{2}\, d + \tfrac{1}{2}'' = 9\tfrac{1}{4} \text{ inches.}$$
$$l = 3\ d = 10\tfrac{1}{2} \text{ inches.}$$

We can use 6 bolts of $\frac{7}{8}$ inch.

The **flange coupling** is shown in Fig. 53. The cast iron

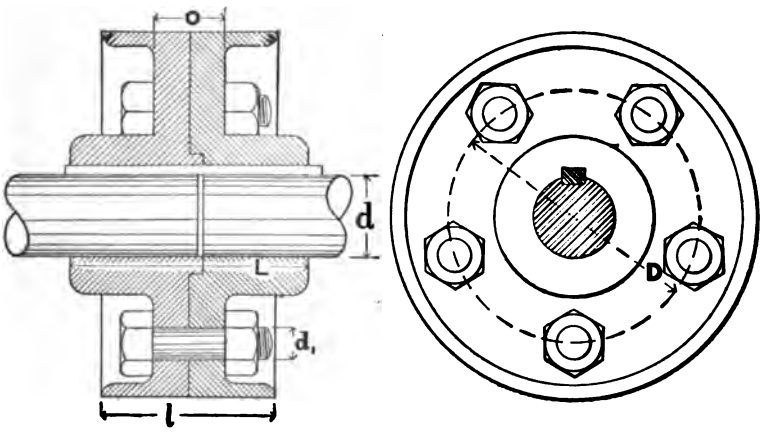


Fig. 53.

flanges are keyed to the ends of the shafts to be connected. The flanges are then brought face to face and bolted together. Sometimes the flanges are faced in a lathe to insure a good joint. One flange is often made to enter the other, as shown in the figure, to prevent the shafts from getting out of line. As in the other shaft

couplings, the ends of the shafts may be enlarged for the key way. For designing, d = the diameter of the shaft, which is taken as the unit.

$$D = 2\tfrac{1}{2}\, d + 2''$$
$$l = 2d.$$

Number of bolts, n, $= 3 + \frac{d}{2}$ (the nearest whole number.)

$$d_1 = \frac{d}{n} + \tfrac{1}{4}''$$
$$o = \tfrac{1}{2}\, d + \tfrac{5}{8}''.$$

Propeller shaft coupling. The coupling shown in Fig. 54 is

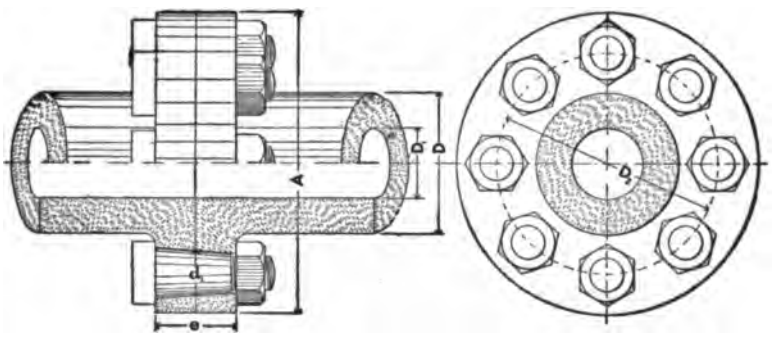
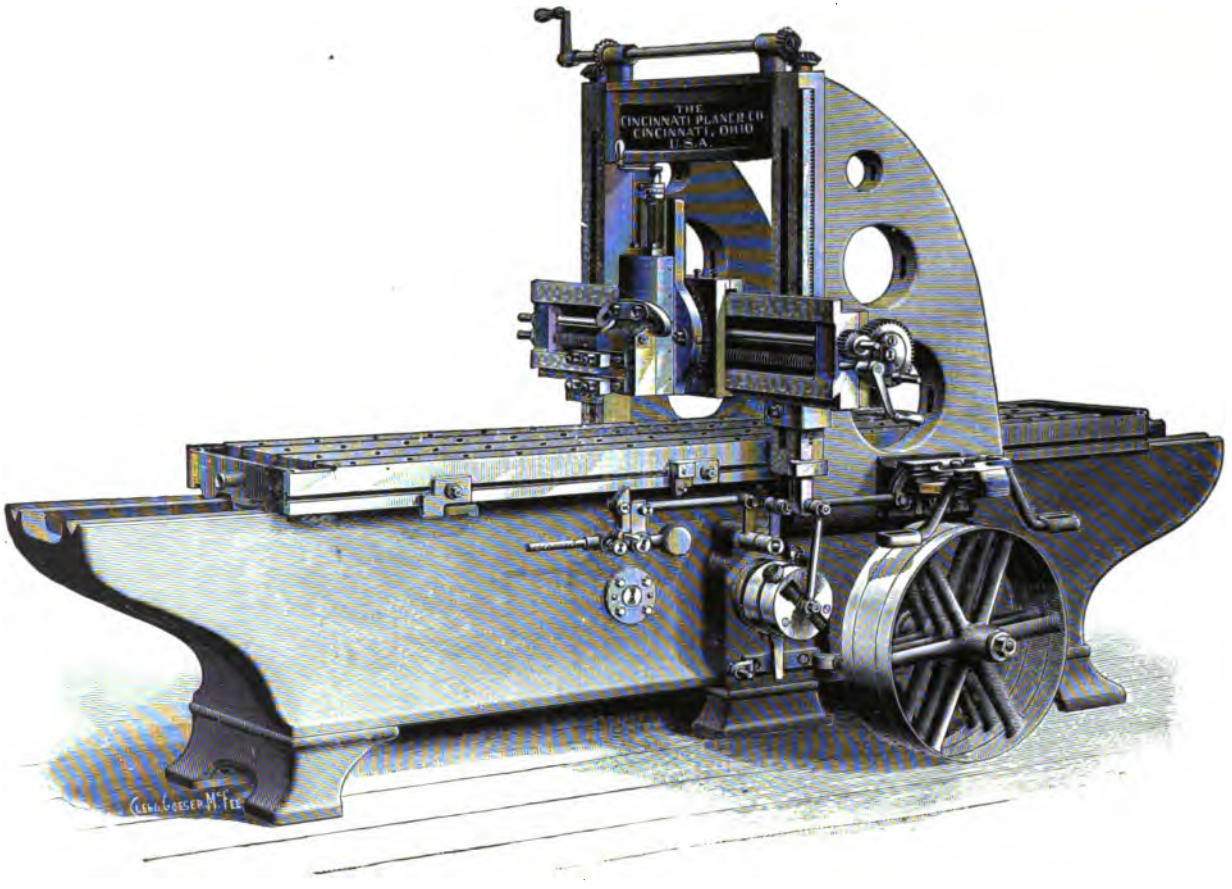


Fig. 54.

used for propeller shafts. In this case the hollow steel shaft is flanged at the ends and joined by bolts. Let d = the diameter of an equivalent solid shaft. Then,

$$d = \sqrt[3]{\frac{D^4 - D_1^4}{D}}$$
$$D_2 = 1\tfrac{3}{4}\, d$$
$$A = 2\, d$$
$$e = \frac{d}{2}$$

The bolts are made tapered. The number of bolts is usually assumed, and the diameter d_1, can be computed by a consideration of the twisting moment of the shaft and the shearing strength of the bolts.



28-INCH PLANER.

Let n = the number of bolts, and a a constant; then $d_1 = a\,d$.

The values of a for different values of n are as follows:

$n =$	3	4	5	6	7	8	9	10
$a =$	.318	.283	.258	.239	.224	.212	.201	.192

Find the dimensions of a shaft coupling for a hollow propeller shaft 4 inches in external diameter and 2 inches in internal diameter using 8 bolts.

$$d = \sqrt[3]{\frac{4^4 - 2^4}{4}} = 3.915 \text{ inches.}$$

$D_2 = 1\frac{3}{4} \times 3.915 = 6\frac{7}{8}$ inches (about).

A = 8 inches (about).

e = 2 inches (about).

$d_1 = .212 \times 3.915 = \frac{13}{16}$.

Bolts $\frac{3}{4}$ inch in diameter would be used.

Sellars Cone Coupling. A convenient coupling for shafts of equal or unequal size is shown in Fig. 55. It consists of an outer

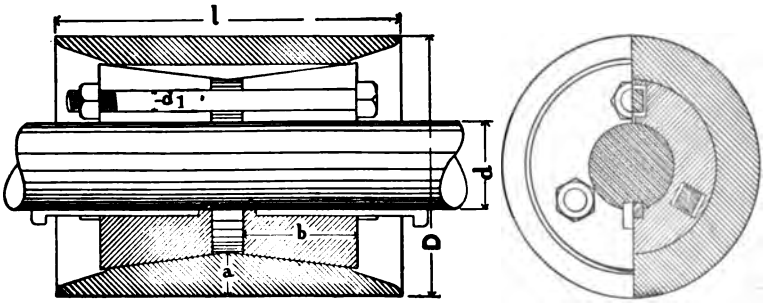


Fig. 55.

cylindrical box or muff which is turned of double conical form on the inside. Two sleeves, the exterior of which are conical and fit the conical surfaces of the box, are placed between the box and the shaft. The inside surfaces of the sleeves fit the shaft. Three square bolts parallel to the shaft and resting in slots cut in the sleeves, press them together. The sleeves are cut through on one side at the bottom of one of the bolt slots. This gives sufficient elasticity so that the sleeves may be drawn inward and grasp the shaft tightly. Each sleeve exerts the same force on the shaft and with the aid of a key prevents slipping. The keys fit at the

sides only. These couplings are easily disconnected if the parts are well oiled before they are put together. The dimensions may be found from the following formulas:

$$D = 3\,d$$
$$l = 4\,d$$
$$b = 1\tfrac{1}{2}\,d$$
$$a = \tfrac{1}{2}\,d$$
$$\text{size of bolts } d_1 = \tfrac{1}{3}d$$

In the above formulas d is the unit, and is taken as the diameter of the shaft. The conical sleeves have a taper of about 4 inches per foot of length.

The Oldham Coupling. Fig. 56 shows a form of coupling used when two shafts are parallel. A disc is keyed on the end of each shaft. Between these discs lies a third which has a

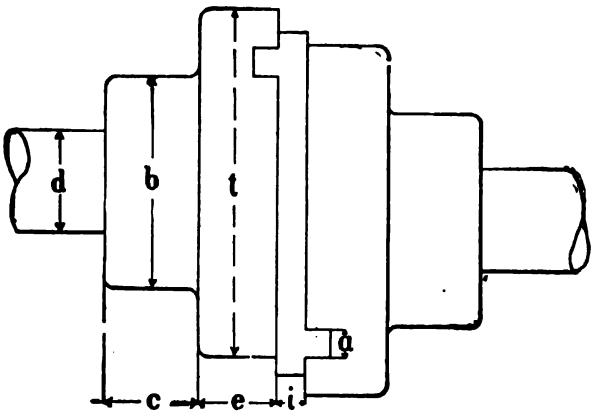


Fig. 56.

feather on each side fitting in a slot in the corresponding disc. The middle disc revolves around an axis parallel to the shafts and midway between them. The shafts and disc have equal velocities.

The proportions for this coupling may be as follows:

d = diameter of shaft,
a = .4 d,
b = 1.75 d,
c = .8 d,
e = .7 d,
i = .25 d,
t = 3 d.

The Universal Coupling. In case two shafts are not in line they may be connected by a universal coupling shown in Fig. 57.

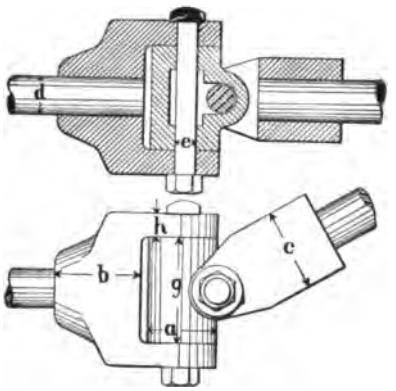


Fig. 57

The velocity ratio varies but little if the angle is small. They are constructed of wrought iron and may have the following proportions; d being the unit.

d = diameter of the shaft,
$a = d$ to $2\ d$,
$b = 1\frac{5}{8}\ d$,
$c = 1\frac{7}{8}$ d,
$e = \frac{1}{2}\ d$,
$g = 2d$,
$h = \frac{5}{8}\ d$.

Loose Couplings are used if shafts are to be connected and disconnected. A type called a claw coupling which somewhat resembles the flange coupling is shown in Fig. 58. It is used for large slow turning shafts, which always revolve in the same direction. This form is easily put in gear. In place of the flanges there is a set of projections or lugs which fit into recesses. One part is firmly keyed to the shaft by a sunk key; the other is fastened by a feather key. The part having the feather key (on the left hand) is prolonged and a groove cut for a lever with which to slide it back and forth.

A coupling which is easily put in gear but can drive only in one direction has its claws shaped as shown in Fig. 59.

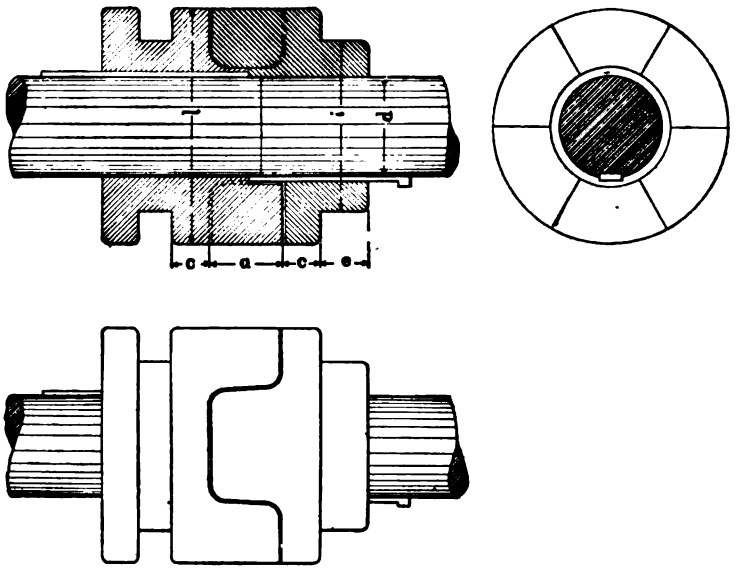


Fig. 58.

The dimensions may be as follows:

d = diameter of shaft,
$a = \frac{5}{8}\ d$,
$c = \frac{3}{8}\ d$,
$e = \frac{1}{8}\ d$,
$i = 1\frac{1}{8}\ d$,
$l = 1\frac{1}{2}\ d$ to $6\ d$.

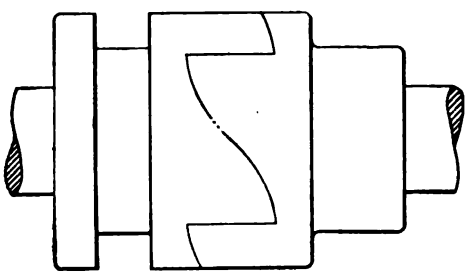


Fig. 59.

Friction Couplings, or clutches serve instead of loose coup-

lings on shafts running at high speeds. Fig. 60 shows a good form of friction clutch. The ring, *e*, is keyed to the shaft *t*. This ring is split and fits inside the cylinder *c*, which is keyed to the shaft *i*. The split ends are connected by a screw having

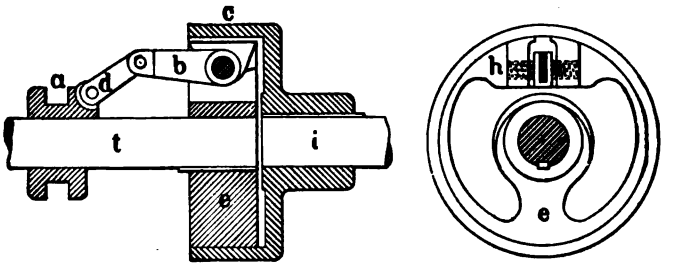


Fig. 60.

right and left hand threads. The link *d*, connects the lever *b*, to the sleeve or collar *a*. The lever *b*, turns the screw. The clutch is readily operated. When the sleeve is pushed toward the cylinder *c*, the rotation of the screw throws the ends *h* of the ring apart, and causes the ring *e* to press firmly against the cylinder *c*.

The proportions for the various parts are about the same as those for Fig. 58. The clutch shown in Fig. 62 is not as good as that of Fig. 60 because it causes an end thrust on the shaft and it is harder to put in gear. It is, however, simple in construction.

Weston Friction Coupling. The friction coupling shown in Fig. 61 is used both as a shaft coupling and for coupling a spur

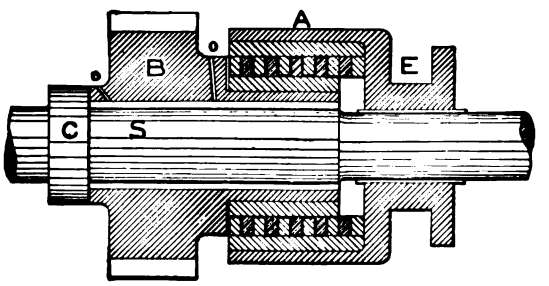


Fig. 61.

wheel to a shaft. The wheel B has a long hub and wrought iron rings or plates which slide on feathers. The clutch box A is fitted on the shaft, slides on feathers, and is moved by a lever working in

the groove E. Inside the box A are six feathers upon which are strung alternately wooden and wrought iron rings. If the coupling box A is pressed to the left there is friction at each face between the rings. The wheel B is prevented from moving endwise by the collar C. One of the great advantages of this coupling is that if a sudden load comes on the spur wheel B, the plates merely slip over each other; if rigidly connected, some part would break.

Fig. 62 shows a simple form of friction coupling or clutch. It is used to couple wheels or pulleys to shafts and for loose

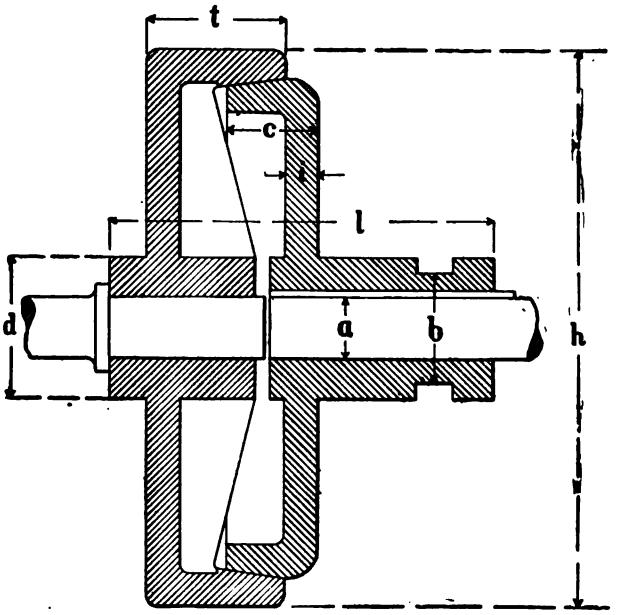


Fig. 62.

couplings for shafts running at high speeds. It consists of a cone keyed rigidly to one shaft and a movable cone sliding on a feather on a second shaft. The movable portion should be placed on the driven shaft so that it will be at rest when out of gear. If the resistance of the driven shaft is considerable the mean cone radius may be three or four times the diameter of the shaft. One great objection to this form is that the horizontal component of the pressure between the conical surfaces causes end thrust on the

shaft. The angle of the cone may be from 4 to 10 degrees. The other proportions are as follows:

a = diameter of shaft,
$b = 1\frac{3}{4}\ a$,
$c = 1\frac{1}{2}\ a$,
$d = 2\ a$,
$h = 4a$ to $8a$,
$i = \frac{3}{8}\ a$,
$t = 2a$.

Shifting Gear for Clutches. Forked levers, having prongs which fit into the groove of the clutch, are used to put clutches in

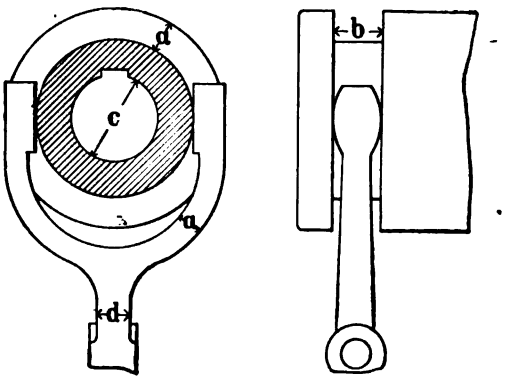
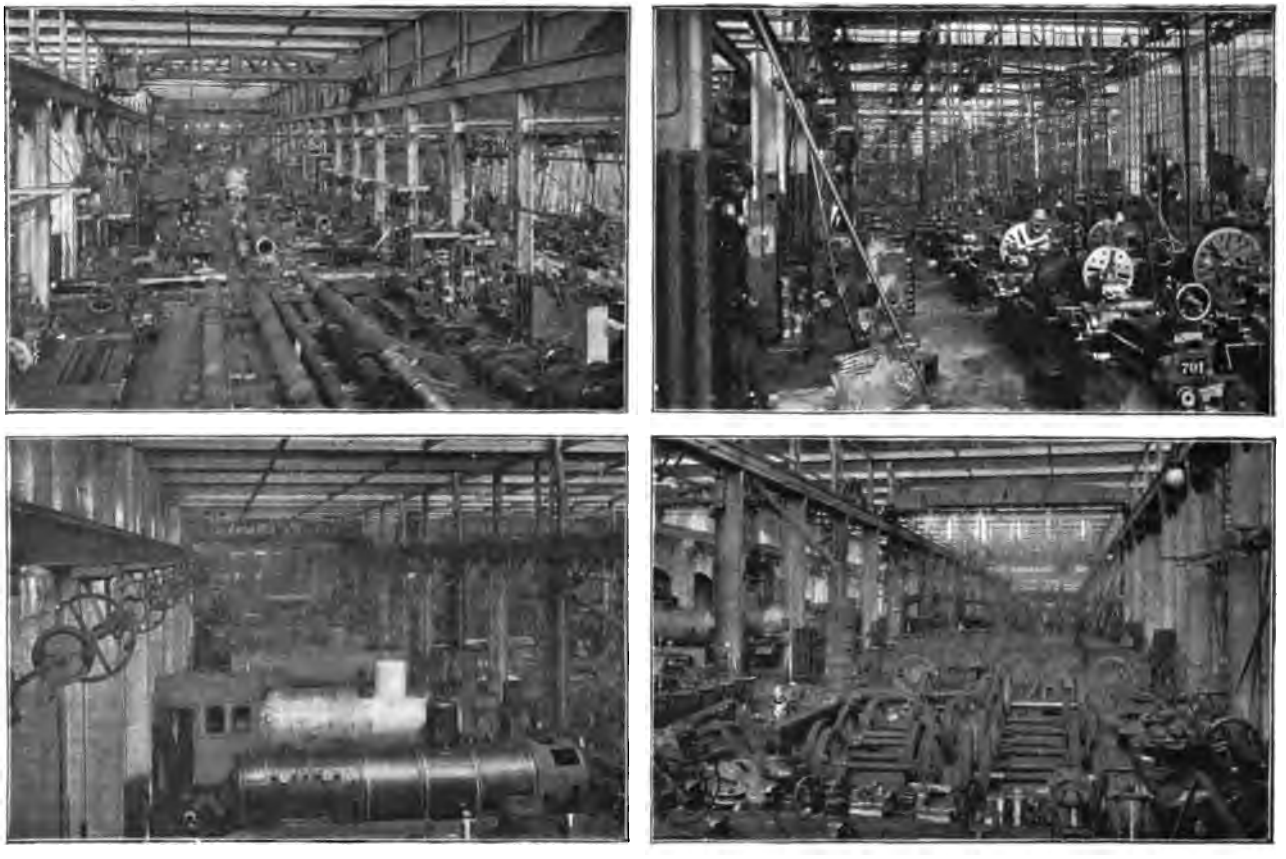


Fig. 63.

and out of gear. The lever is ordinarily worked by hand. Sometimes a brass strap is made to encircle the groove; this increases the wearing surface. The following dimensions refer to Fig. 63; the unit being the diameter of the shaft.

c = the diameter of the shaft,
$a = \frac{5}{16}\ c$,
$b = \frac{1}{2}\ c$,
$d = \frac{3}{8}\ c$.



INTERIOR VIEWS OF A REPRESENTATIVE EUROPEAN LOCOMOTIVE SHOP

(By courtesy of the *Engineering Magazine*, New York and London)

SECTION 2.

MACHINE DESIGN.

PART II.

BEARINGS.

A simple form of journal bearing used for rough work, is a cylindrical hole in some portion of the frame which supports the shaft. In time, this hole wears oval and as there is no means of adjustment it is undesirable. In case such a bearing must be used, the hole should be bored larger than the journal and a **lining** or **bushing**, made of brass or other soft metal, placed around the journal. When the lining wears, it may be replaced with little trouble. The journal, being of harder material than the lining, wears but little.

It is often undesirable and sometimes impossible to use solid bearings. The bearing is then divided, and the parts held together by bolts. The parts of the lining are called **steps** or **seats**. The bushing is divided so that it can be adjusted from time to time; when the wear is considerable new bushing is inserted.

The method of dividing the lining or brasses, as they are often called, depends upon the direction of pressure. Pressure generally acts *constantly* in one direction as in the case of the shaft of a factory engine; or with *alternating* direction as in the case of the crank-pin of an engine. The bearing should be so divided that the plane of division is perpendicular to the pressures, or it should be in the direction in which the load is least.

Fig. 1 represents some of the common ways of dividing bushings. In A, B, and C the direction of the load is vertical and so the seats are divided horizontally. The bearing surfaces are cylindrical while the outside may have various shapes corresponding to the shape of the pedestal. In D the direction is oblique and the bearing is divided as shown. The method of division shown at E is used when the pressure is oblique but the adjustment can

be most conveniently made horizontally. F shows a form sometimes used when the direction varies and it is necessary to use several steps.

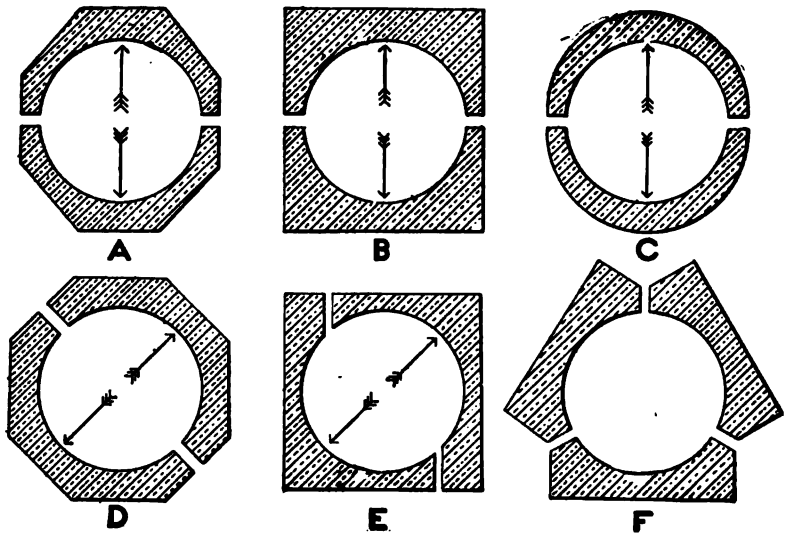


Fig. 1.

A simple form of bearing is shown in Fig. 2. It is of cast iron being a part of the frame, but it is divided to allow adjustment. As the hole becomes oval, the wear being at the top and bottom, the top is set by means of the bolts until the hole becomes circular. Sometimes grooves are cut in it which are filled with babbitt metal. The proportions given in Fig. 2 are decimals of the diameter.

It is usual to line bearings with seats or steps of brass, gun-metal, babbitt metal, phosphor-bronze, or some other alloy. The interior of the steps is made to fit the journal while the exterior may be of some other shape. When made octagonal, each step fits into an octagonal recess in the pillow-block which prevents its turning. An easier method is to bore the pedestal and turn the lining to fit. If this is done the seats must be provided with pins or lugs to prevent them from turning.

Steps are sometimes fitted only at the ends; the intermediate part being left rough.

Fig. 3 shows an ordinary seat having the outside shaped to fit an octagonal recess. Seats may be made from the following proportions, d being the unit.

d = diameter of journal	$e = .1\ d$
$a = .1\ d + \frac{3}{16}''$	$h = \frac{1}{8}\ d + \frac{1}{8}''$
$b = \frac{1}{8}\ d$	$t = \frac{1}{4}\ d$
$c = \frac{1}{8}\ d + \frac{3}{16}''$	D $= 1\frac{1}{2}\ d$

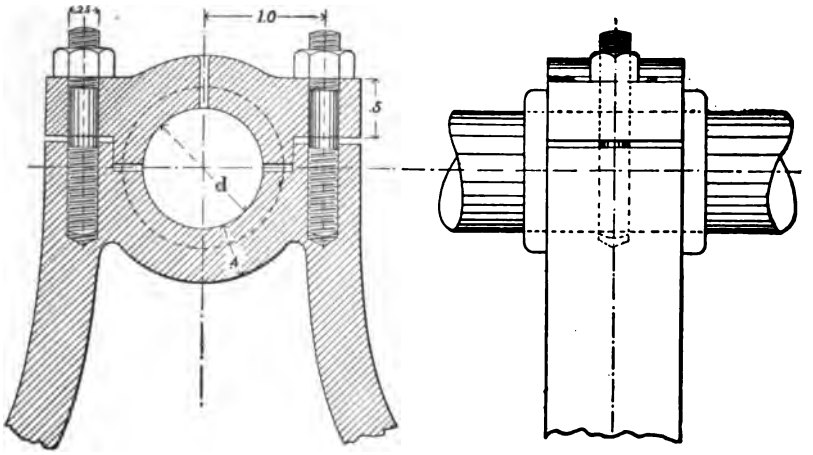


Fig. 2.

The thickness of brass at a is made a little larger than at e because the wear is greater at that point. The dimension o is determined from the pillow-block.

In order that the journal may be well lubricated, the seats should be provided with grooves or channels for oil. These grooves are made parallel to the axis of the shaft, at the top, so that the oil may readily flow to them. The advancing side of the journal will carry a thin film of oil around with it. Grooves vary from $\frac{1}{8}''$ to $\frac{1}{2}''$ in width according to the diameter of the shaft.

Linings. It is common practice to line the seats with some anti-friction metal. Babbitt metal is one of the most used. Bearings, having such a lining, run cooler than when made entirely of gun-metal or brass. The reason for this is probably due to the ease with which the softer metal wears and accommodates itself to the journal.

A few common methods for lining seats are shown in Fig. 4. At A the babbitt metal is shown cast in a series of shallow circular holes; in the one represented at B, the helical grooves take the place of the circular holes and at C the shallow rectangular grooves are cut in the step to receive the babbitt metal. Bearings so lined wear better than when having a complete lining of babbitt metal.

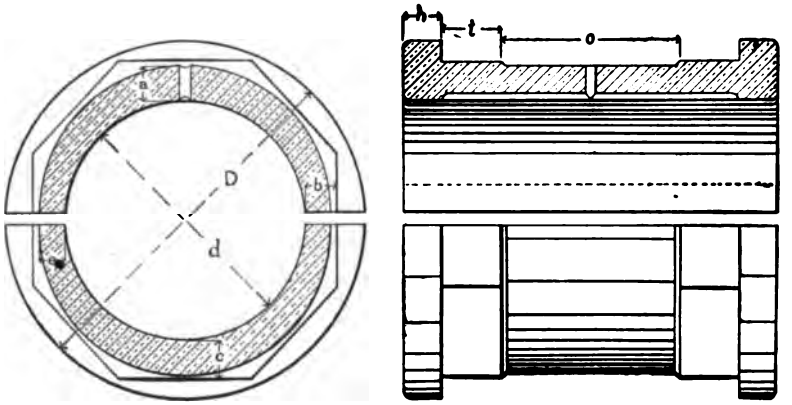


Fig. 3.

Frequently the seats are made entirely of babbitt metal because so easily made. The melted babbitt metal is poured around a mandrel which is the exact size of the journal and placed in the proper position in the bearing. For a better grade of work

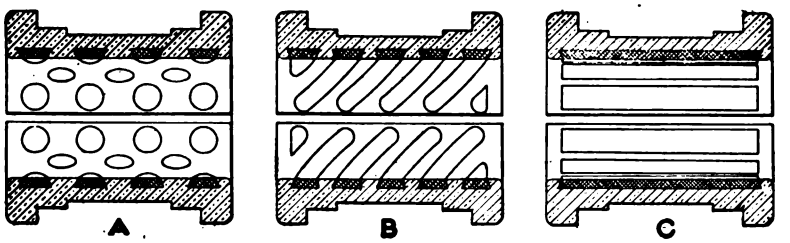


Fig. 4.

a smaller mandrel is used; when the metal becomes hard it is hammered into the seat and then bored the exact size.

This process of lining the seats is called **babbitting** and the bearing is said to be babbitted.

PEDESTALS.

A form of bearing that supports a journal is called a **pedestal.** It is also called a **pillow-block, bearing** or **journal-box.**

For small journals, that is, those having a diameter of less than 2 inches, a pedestal like that shown in Fig. 5 is convenient, neat and compact. The two principal parts are the base, or box, which supports the journal, and the cap which is held to the base by tap bolts or cap screws. The seats of this form of journal bearing are of babbitt metal. The proportions of all pedestals are determined largely by experience, no part being designed by calculation.

For the pedestal shown in Fig. 5, d is the diameter of the journal and is taken as the unit for all proportions.

d = diameter of shaft.
$a = d$
$b = .25\ d$ to $.375\ d$
$c = 1.75\ d$
$e = 2.25\ d$ to $2.5\ d$
$f = .25\ d$
$l = 2.5\ d$
$s = 1.75\ d$

For larger journals a more elaborate form of pedestal is used. Fig. 5 on page 21 of "Machine Design" Part I. shows three views of a pedestal for a three-inch journal. All the dimensions are given in inches and fractions of inches, also the finish marks and names of the materials. The pedestal shown in Fig. 6 is suitable for almost any size of journal. The lowest portion, called the foundation plate may be omitted. Above this is the block which supports the seats and journal. In general it is like that shown in Fig. 5. The steps are octagonal and have flanges to prevent lateral movement. The cap is held to the block by two bolts which have lock nuts. The under surface of the block and the top surface of the foundation plate have narrow chipping strips so that the level may be easily adjusted. The bolt holes in both the block and in the foundation plate are oblong in order that the position of the block may be changed slightly. When adjusted the block is kept in position by hard wood or iron wedges driven into the

space between the ends of the blocks and the jaws cast on the foundation plate.

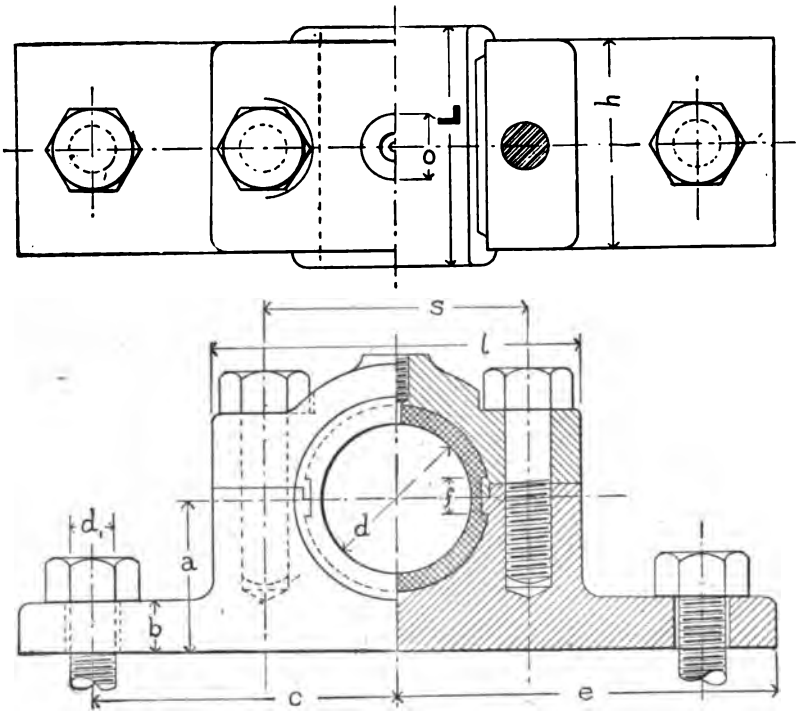


Fig. 5.

The proportions, expressed in terms of d, the diameter of the journal, are as follows:

a	$= \frac{1}{2}\,d + \frac{1}{8}''$	m	$= \frac{5}{8}\,d$ to $\frac{3}{4}\,d$
b	$= d$ to $1\frac{1}{4}\,d$	n	$= \frac{3}{8}\,d$
c	$= 1\frac{3}{4}\,d$ to $2\,d$	o	$= 1\frac{5}{8}\,d$
d_1	$= \frac{1}{4}\,d$	r	$= \frac{1}{4}\,d$ to $\frac{1}{2}\,d$
e	$= 1\frac{1}{8}\,d$	s	$= \frac{1}{4}\,d$
i	$= \frac{1}{16}\,d$ to $\frac{1}{8}\,d$	t	$= \frac{5}{16}\,d$
l	$= 1.5\,d$ to $2\,d$	$2\,x$	$= \frac{1}{2}\,d$
		$2\,y$	$= \frac{5}{8}\,d$ to $\frac{7}{8}\,d$

Ball and Socket Bearing. If the steps of a pedestal are of cast iron they must be long to give sufficient bearing surface. The cast iron answers as well as brass or babbitting if the steps are long

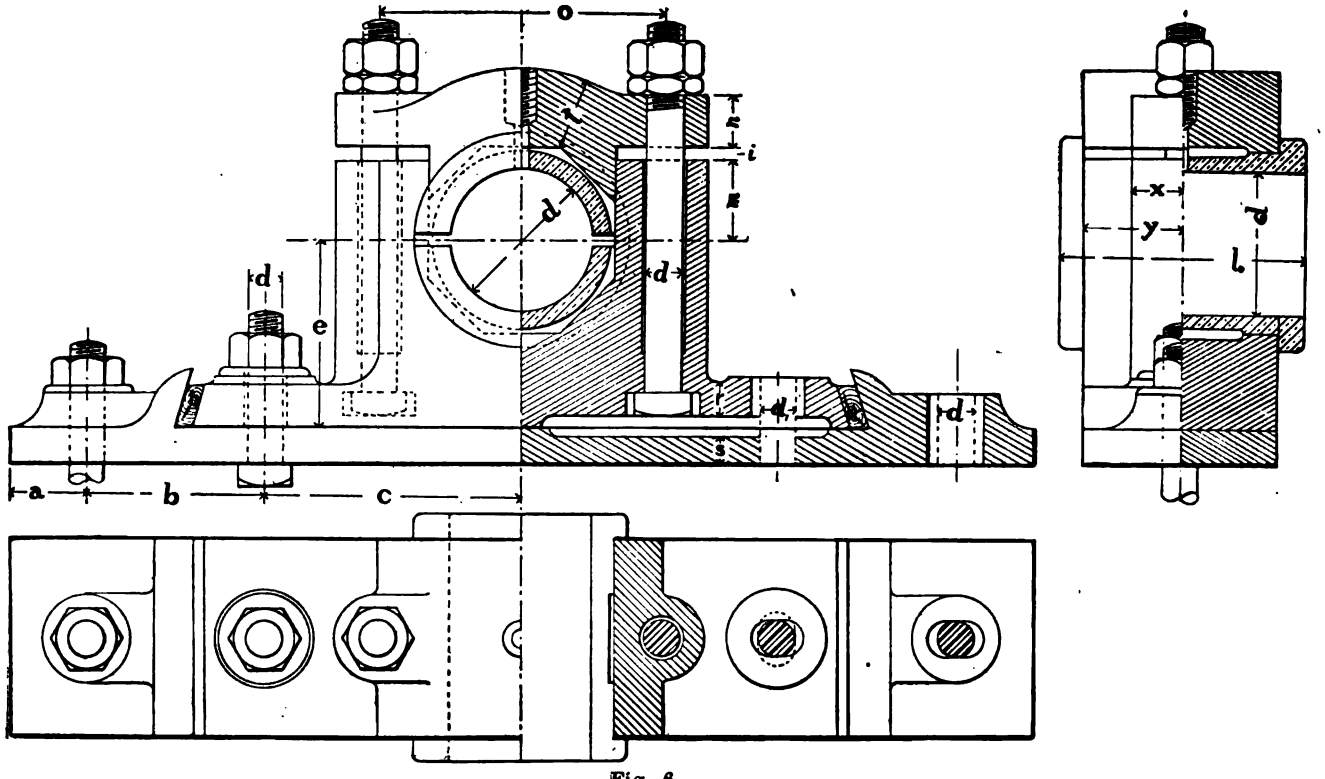


Fig. 6.

enough. The longer the bearings the more difficult it is to make them exactly concentric and in line. For long bearings, the seats are made with a spherical enlargement at the center which fits into corresponding hollows in the cap and blocks. This makes them self-adjusting, and allows the shaft to move slightly in any direction according to wear or imperfect alignment of the shaft. Fig. 7 shows two views of a ball and socket bearing invented by Mr. Sellars. It is designed for fast running shafts and has long cast iron seats. These bearings are supported on spherical seats for ease of adjustment. The bearing is lubricated by an oil cup or

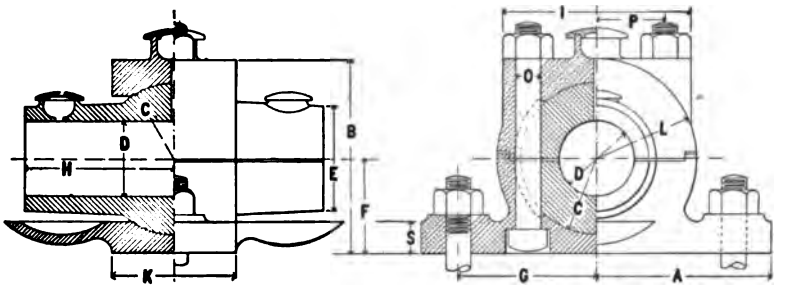


Fig. 7.

reservoir at the center of the cap. On account of the length, two cups filled with a mixture of tallow and oil are placed at each end. The oil and tallow are solid at ordinary temperatures but melt at about 100° F. If the central oil cup fails to provide lubricant enough to prevent heating, the high temperature of the bearing causes the mixture to melt and prevent injury. The lower portion of the bearing is formed at each end into a drip cup. The dimensions may be taken from the following proportions which are in terms of the diameter of the journal.

$A = 2\frac{1}{4}\ d$ to $2\frac{3}{4}\ d$ $\quad$ $H = 2\ d$ to $3\ d$

$B = 2\frac{1}{2}\ d$ to $2\frac{5}{8}\ d$ $\quad$ $I = 2\frac{1}{2}\ d$ to $3\frac{1}{2}\ d$

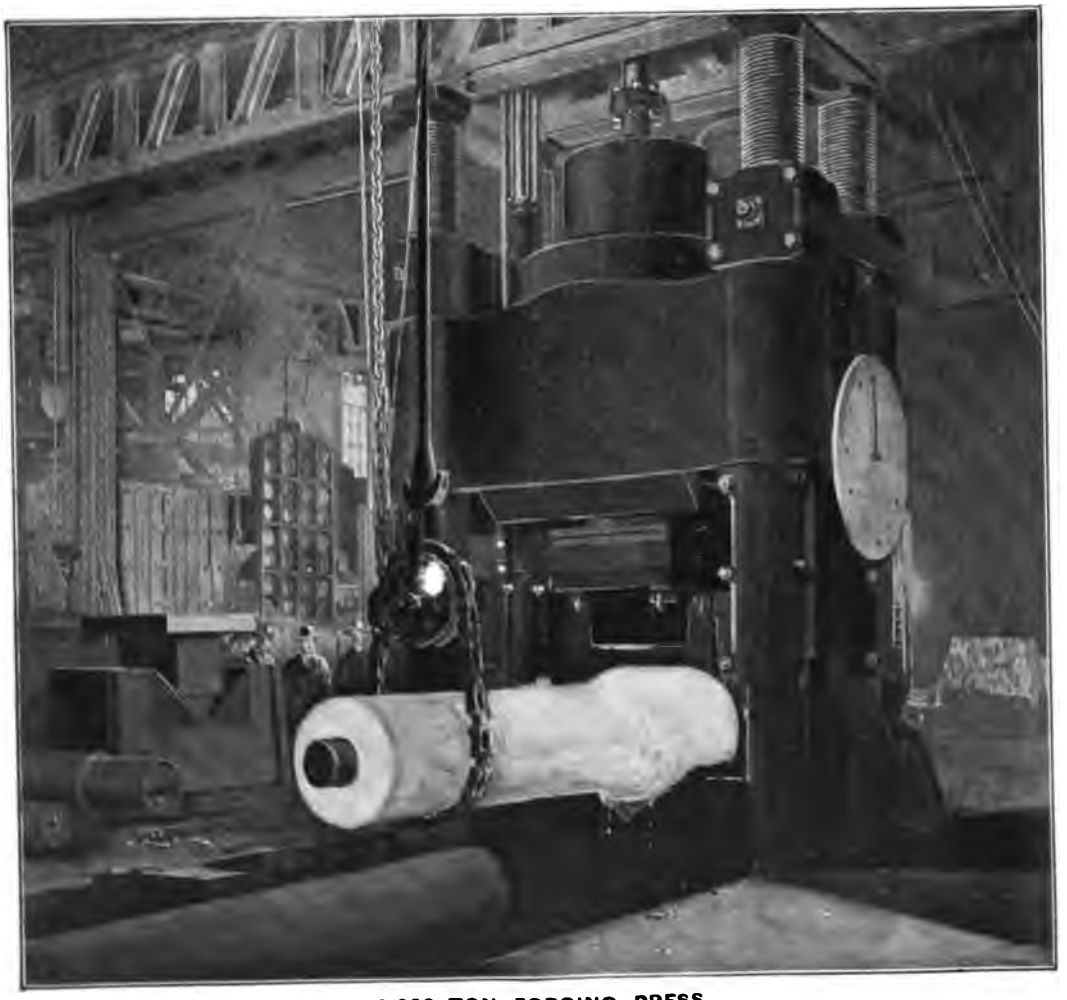
$C = \frac{7}{8} d$ $\quad$ $K = 1\frac{3}{4}\ d$

$E = 1\frac{1}{4}\ d + \frac{1}{2}''$ $\quad$ $L = 1\frac{3}{8}\ d$

$F = 1\frac{3}{8}\ d$ $\quad$ $O = \frac{1}{4}\ d + \frac{1}{4}''$

$G = 1\frac{3}{4}\ d$ $\quad$ $P = \frac{3}{4}\ d$ to d

$S = \frac{3}{8}\ d$



9,000-TON FORGING PRESS

The direction of the resultant of the pressure on bearings for crank shafts is oblique because the load is due both to the weight of the shaft and fly wheel and to the alternate push and pull of the connecting rod. For this reason the division of the pedestal is often oblique.

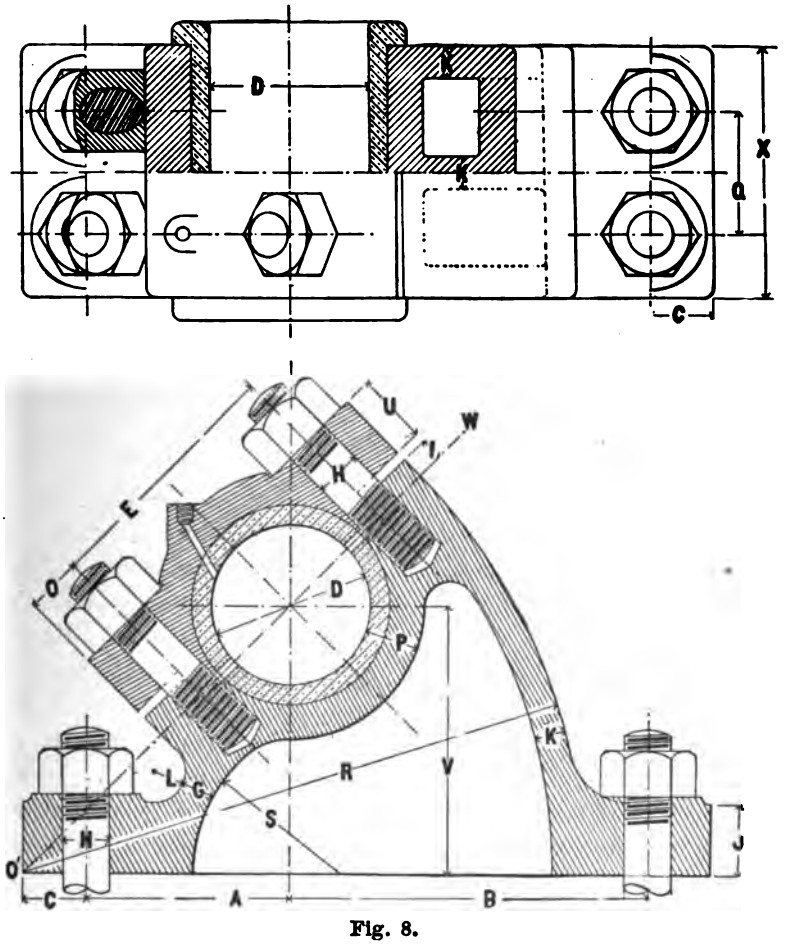


Fig. 8.

Fig. 8 shows a pedestal of this form, the division being at an angle of 45 degrees to the horizontal. The two views show the construction so that a description is unnecessary. The proportions are as follows, d being the unit.

$A = 1\frac{1}{8}\ d$	$K = \frac{1}{8}\ d + \frac{1}{4}''$
$B = 2\ d + \frac{3}{4}''$	$O = \frac{3}{8}\ d$
$C = \frac{1}{4}\ d + \frac{1}{2}''$	$P = \frac{3}{8}\ d$
$E = 1\frac{3}{4}\ d$	$Q = \frac{3}{4}\ d$
$G = \frac{1}{8}\ d + \frac{1}{4}''$	$S = \frac{3}{4}\ d$ to $1\frac{1}{4}\ d$
$H = \frac{1}{8}\ d + \frac{3}{16}''$	$U = \frac{1}{2}\ d$
$I = \frac{3}{16}\ d$	$V = 1\frac{1}{2}\ d$
$J = \frac{1}{4}\ d + \frac{1}{2}''$	$X = 1\frac{1}{2}\ d$

The radius R is found by trial. The center may be taken at O′ as shown in the figure or it may be the point where the line O′ W intersects a line parallel to the base line at a distance J above it. The radius L is found by trial. In case the bearing is long it may be tapped for two oil cups as shown.

Pivot or Foot-step Bearings. The pedestal in which a vertical shaft rests is called a foot-step or pivot bearing. The usual form is shown in Fig. 9. The steel end of the shaft rotates on a flat or slightly hollowed disc. The disc is either of steel, brass, or bronze. Lateral movement is prevented by a brass bush fitted in the cast iron pedestal. In case the downward pressure is great, the bush may be prevented from splitting by having a wrought iron ring shrunk on it. The unit for the proportions is d, the diameter of the pivot.

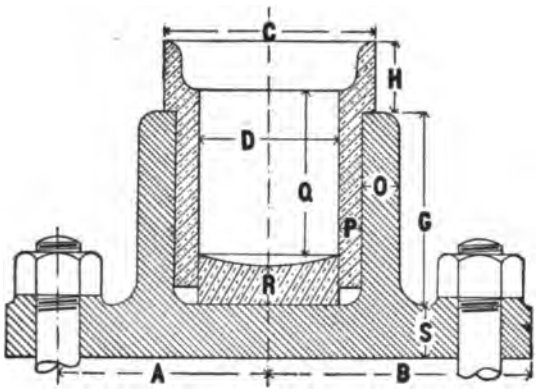


Fig. 9.

$A = 1\frac{1}{2}\ d$	$O = \frac{1}{4} d$
$B = 2\ d$	$P = \frac{1}{8}\ d + \frac{1}{8}''$
$C = 1\frac{1}{2}\ d$	$Q = 1\frac{1}{4}\ d$
$G = 1\frac{1}{2}\ d$	$R = \frac{1}{4}\ d + \frac{1}{8}''$
$H = \frac{1}{2} d$	$S = \frac{3}{8}\ d + \frac{1}{4}''$

Pivots with Loose Discs. If a pivot runs at high speed the relative speed of the surfaces in contact will be high. To reduce

this speed a number of loose discs of gun metal or bronze are placed between the pivot and the foot-step. This construction is shown in Fig. 10. The discs distribute the pressure and wear, and if any one surface is temporarily imperfectly lubricated the rubbing takes place at the other surfaces. The pivots and discs are lubricated by means of a channel.

The proportions are:

$a = \frac{3}{16}\ d$

d = diameter of pivot

$c = \frac{1}{4}$ to $\frac{3}{8}\ d$

To find the number of discs:

Let n = the number of revolutions per minute of the pivot.

D = the number of discs.

Then the relative speed of the surfaces in contact is,

$\frac{n}{D+1}$ revolutions per minute.

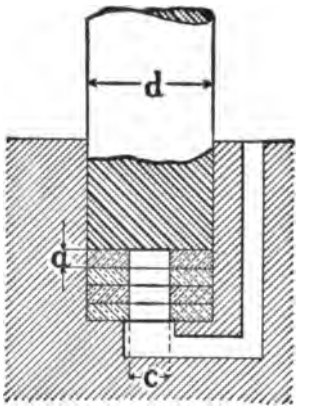


Fig. 10.

Suppose we have a pivot running at 2,500 revolutions per minute and we wish to reduce the speed of the adjacent surfaces to 625 revolutions.

$$625 = \frac{2{,}500}{D+1}$$

or, $D = 3$.

Hence 3 discs would be used.

Exact vertical and lateral adjustment are often necessary. Fig. 11 shows an arrangement by which the vertical adjustment is obtained with the use of a set screw and lateral adjustment by four set screws. These screws have lock nuts to keep the brasses in position when once adjusted.

The unit d for Fig. 11 is $D + \frac{1}{2}''$.

$K = \frac{3}{4}d$ to $\frac{7}{8}\ d$

$O = d + 1\frac{1}{8}''$.

$Q = 1\frac{1}{2}\ d$.

WALL BRACKETS AND HANGERS.

Wall Brackets. It often happens that shafts must be supported by a bearing fixed to the wall of the factory. For such

cases a wall bracket is used. A wall bracket is a cantilever with a uniform load due to its own weight and a concentrated load at the end due to the weight and the downward pull of the shaft. A mathematical treatment of the subject, shows that a cantilever of constant width and uniformly strong at all sections should have a parabolic form for a concentrated load at the end and a triangular form for a uniform load.

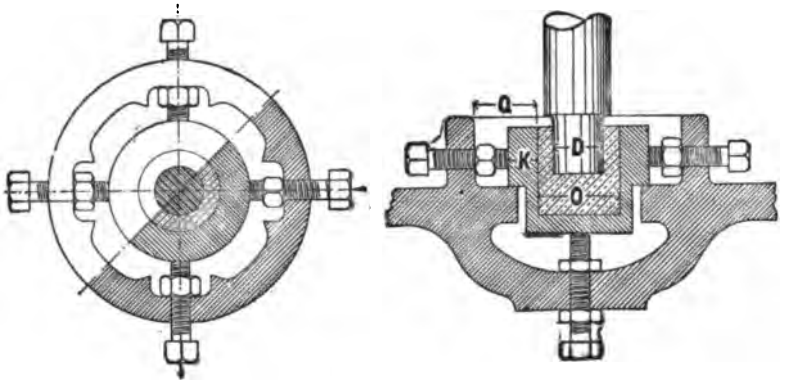


Fig. 11.

In practice the form of a bracket is neither parabolic nor triangular but approximates both. If convenient the brackets are made short in order to reduce the bending moment.

Fig. 12 shows a good simple form of bracket. The section of the frame is similar to an I beam section. A tin dish may be placed in the recess under the bearing, to catch oil drippings.

The unit d for this bracket is D + $\frac{1}{2}''$; the proportions being as follows:

A = 5 d to 6 d	K = $1\frac{1}{2}d$
B = 7 d to 8 d	L = 8 d
C = 6 d	O = $\frac{3}{4}$ d to $\frac{7}{8}$ d
E = $1\frac{1}{2}d$	P = $1\frac{3}{4}$ d
G = $1\frac{1}{4}d$	R = $1\frac{7}{8}d$
H = $\frac{7}{8}$ d	S = $1\frac{5}{8}d$
I = $\frac{1}{4}$ d = $\frac{1}{2}''$	U = $1\frac{1}{2}d$
J = $\frac{5}{16}$ d	V = $\frac{7}{8}$ d
	W = $1\frac{1}{2}d$ to 8 d

It is often more convenient or less expensive to place an ordinary pedestal on a bracket as shown in Fig. 13. The unit

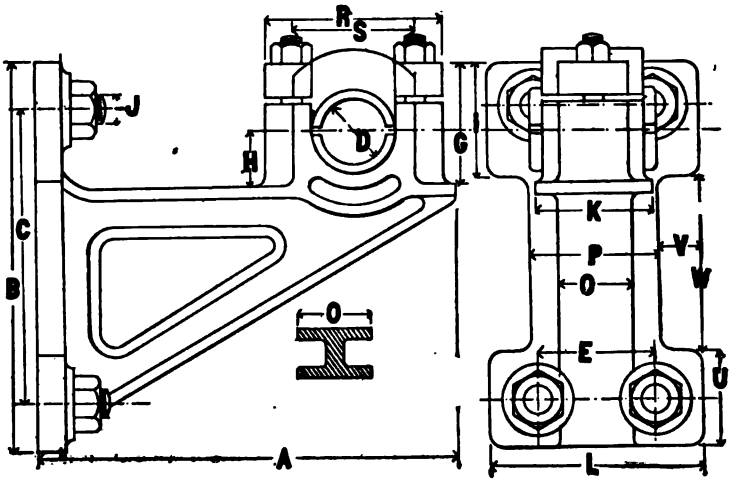


Fig. 12.

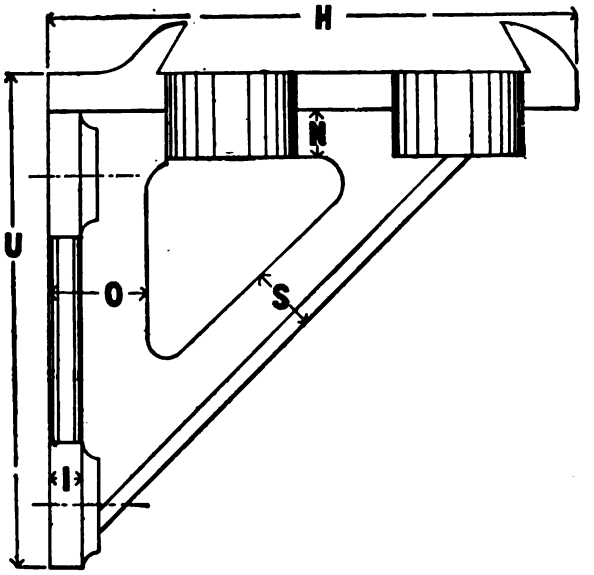


Fig. 13.

for the bracket may be D, the diameter of the shaft; the other proportions in terms of D are as follows:

$H = 7\ D$ $\quad N = \frac{3}{4}\ D$
$I = \frac{1}{2}\ D$ $\quad U = 6\ D$ to $7\ D$
$O = \frac{7}{8}\ D$ $\quad S = \frac{3}{4}\ D$

There are many varieties of hangers made by the various manufacturing companies; some have long bearings and spherical seats so that they are self-adjusting.

Hangers. Shaft bearings are suspended from the ceiling by hangers. The pedestal for hangers is modified somewhat. There are many forms of hangers; some are fastened to the ceiling by both sides and others by but one side. The pedestal may be designed by the ordinary proportions already given.

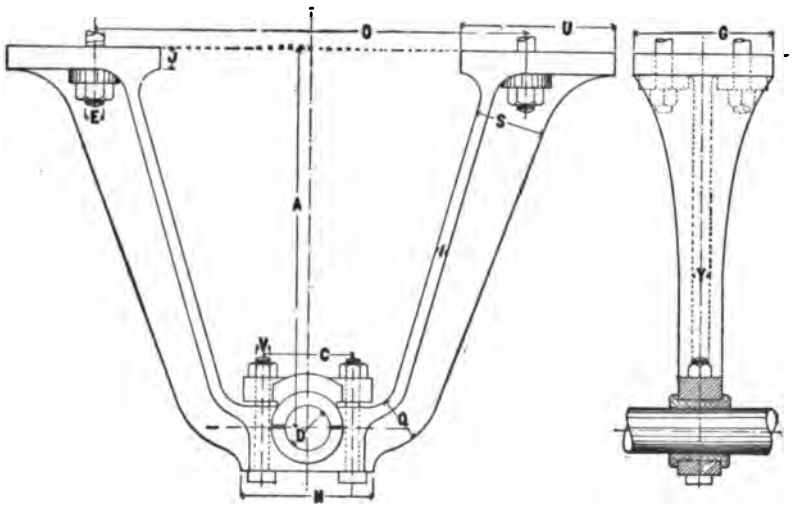


Fig. 14.

In the pedestal shown in Fig. 14, the pedestal frame is bent round and upward and fastened to the wall on both sides.

The unit $d = D + \frac{1}{2}''$.

The proportions for Fig. 14 are

$A = 5\ d$ to $7\ d$ $\quad O = 10\ d$
$C = 1\frac{3}{4} d$ $\quad Q = d$
$E = \frac{1}{4}\ d$ $\quad S = 1\frac{1}{2} d$
$G = 2\ d$ to $2\ \frac{1}{2}\ d$ $\quad U = 3\ d$
$I = \frac{1}{8}\ d + \frac{1}{8}''$ $\quad V = \frac{1}{4}$ d
$J = \frac{3}{8}\ d$ $\quad Y = \frac{1}{8}\ d + \frac{1}{8}''$
$N = 2\frac{1}{4}\ d$

The bearing of the hanger shown in Fig. 15 is similar to that shown in Fig. 7. The lower frame is divided, so that the shaft may be easily removed. The parts are connected by bolts. The steps are of cast iron without a lining of brass or babbitt metal and are bored to fit the shaft.

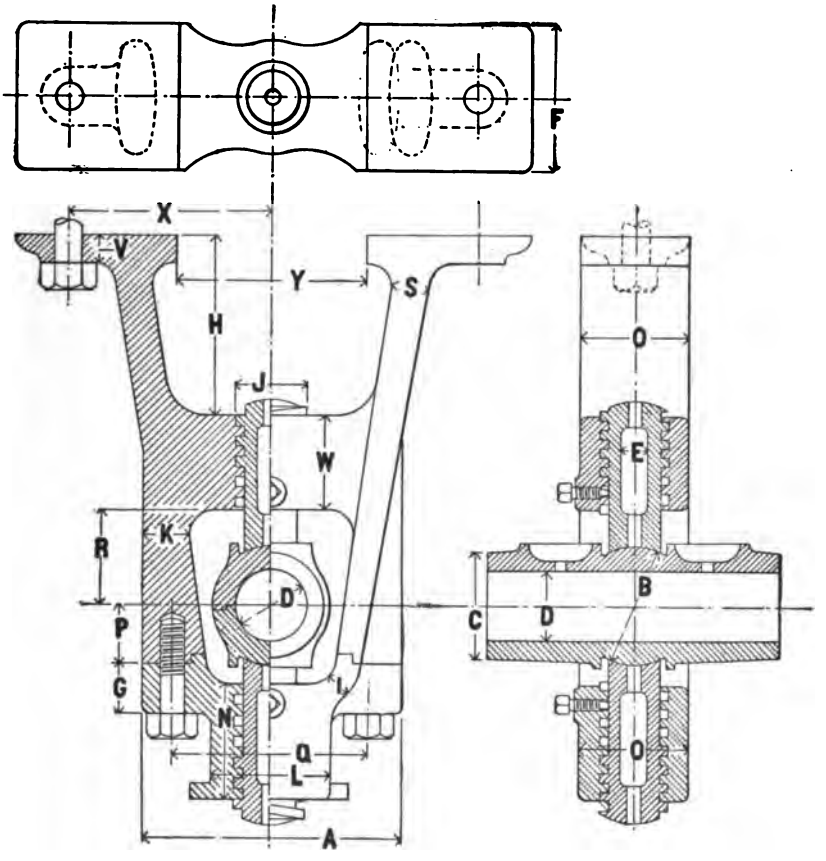


Fig. 15.

The drop D′ (the distance between the center of the shaft and the top of the hanger) is adjusted by the plugs which are screwed up and down by a square key inserted at the ends. Set screws are used to lock the plugs in the desired positions.

The proportions for this hanger are in terms of D, the diameter of the shaft and D′ the drop.

$A = 3\frac{1}{2}D$	$L = 1\frac{3}{4}D$
$B = 1\frac{3}{8}D + \frac{3}{8}''$	$N = 1\frac{1}{2}D$
$C = 1\frac{1}{4}D + \frac{1}{4}''$	$O = 1\frac{1}{8}D + \frac{1}{2}''$
D = diameter of shaft	$P = \frac{7}{8}D$
$E = \frac{3}{8}D$	$Q = 2\frac{1}{2}D$
$F = 2D$	$R = 1\frac{1}{4}D$
$G = \frac{3}{4}D$	$S = \frac{1}{2}D$
$H = \frac{1}{2}D'$	$V = \frac{1}{2}D$
$I = \frac{1}{4}D + \frac{1}{8}''$	$W = \frac{1}{4}D'$
$J = \frac{15}{16}D$	$X = 2D + \frac{3}{16}D'$
$K = \frac{5}{8}D$	$Y = D + \frac{1}{8}D'$

Trays to catch oil drippings may be hung to the projections at the lower end of the hanger. The plugs usually have two

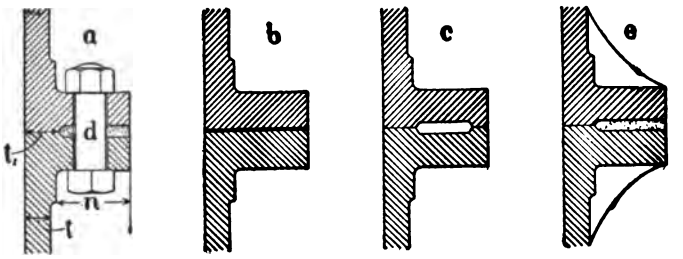


Fig. 16.

threads to the inch, that is, $\frac{1}{2}$ inch pitch. It is better to have the threads on the plugs made in a lathe than to have them cast on the plugs.

FLANGES.

Flanges and bolts are commonly used for joining ends of cast iron pipes and elbows, as well as cast iron plates. Where flanged plates are to be connected the flanges are cast on the plates; they may be external or internal. The thickness of the flanges is greater than the plates, and the bolts for ordinary work are $\frac{3}{4}$ inch or more in diameter.

The common arrangements of bolts and flanges are shown by Fig. 16. An ordinary **rust joint** is shown at *a*. Its flanges fit

at a chipping strip, t, the rest of the space is filled with iron cement. b is a **faced joint** with some kind of packing ring. The flange at c has two faced portions so that the bolts cannot bend the flanges.

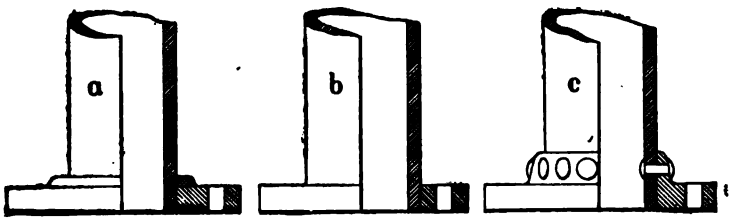


Fig. 17.

Feathers or brackets between the bolts strengthen the one shown at e. The bolt diameter, d, is found from the equation:

$d = \frac{5}{4}\,t + \frac{1}{8}''$ to $\frac{3}{4}\,t + \frac{1}{2}''$ and $t_1 = .9\,t + \frac{3}{8}''$

The distance between bolts is 6 d or less if necessary for strength, and the width of flange n is $2\frac{1}{4}\,d + \frac{1}{4}''$

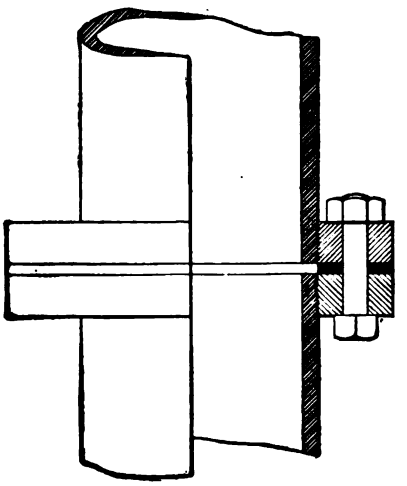


Fig. 18.

The larger sizes of cast-iron pipe, especially where high pressures are used, should be joined by flanges. Fig. 17 shows three forms of pipe flange joints. The flange at a is of cast-iron and is screwed to the end of the pipe. Such a flange is used for sizes of pipes less than 6″. b is a wrought iron or steel flange with the end of the pipe slightly riveted. For large pipes and high pressures the form c is used. The flange and pipe are riveted to give great strength.

The following is a table of "Standard Pipe Flanges" adopted by the American Society of Mechanical Engineers.

STANDARD PIPE FLANGES.

Pipe Diameter, (ins.)	Pipe Thickness, (ins.)	Stress on Pipe per sq. in. at 200 lbs.	Flange Diameter, (ins.)	Flange Thickness, (ins.)	Width Flange Face, (ins.)	Bolt Circle Diameter, (ins.)	Number of Bolts.	Bolt Diameter, (ins.)	Bolt Length (ins.)	Stress on each bolt per sq. in. at bottom of thread at 200 lbs.
2	.409	460	6	$\frac{5}{8}$	2	$4\frac{3}{4}$	4	$\frac{5}{8}$	2	825
$2\frac{1}{2}$	.429	550	7	$\frac{11}{16}$	$2\frac{1}{8}$	$5\frac{1}{4}$	4	$\frac{5}{8}$	$2\frac{1}{4}$	1050
3	.448	690	$7\frac{1}{2}$	$\frac{3}{4}$	$2\frac{1}{4}$	6	4	$\frac{5}{8}$	$2\frac{1}{2}$	1330
$3\frac{1}{2}$	.466	700	$8\frac{1}{2}$	$\frac{13}{16}$	$2\frac{1}{4}$	$6\frac{1}{2}$	4	$\frac{5}{8}$	$2\frac{1}{2}$	2530
4	.486	800	9	$\frac{15}{16}$	$2\frac{3}{8}$	$7\frac{1}{4}$	4	$\frac{5}{8}$	$2\frac{3}{4}$	2100
$4\frac{1}{2}$	.498	900	$9\frac{1}{4}$	$\frac{15}{16}$	$2\frac{3}{8}$	$7\frac{3}{4}$	8	$\frac{3}{4}$	3	1430
5	.525	1000	10	$\frac{15}{16}$	$2\frac{1}{2}$	$8\frac{1}{2}$	8	$\frac{3}{4}$	3	1630
6	.563	1060	11	1	$2\frac{5}{8}$	$9\frac{5}{8}$	8	$\frac{3}{4}$	3	2360
7	.60	1120	$12\frac{1}{2}$	$1\frac{1}{16}$	$2\frac{5}{8}$	$10\frac{3}{4}$	8	$\frac{3}{4}$	$3\frac{1}{4}$	3200
8	.639	1280	$13\frac{1}{2}$	$1\frac{1}{8}$	$2\frac{3}{4}$	$11\frac{3}{4}$	8	$\frac{3}{4}$	$3\frac{1}{2}$	4190
9	.678	1310	15	$1\frac{1}{8}$	$2\frac{7}{8}$	13	12	$\frac{3}{4}$	$3\frac{1}{2}$	3610
10	.713	1330	16	$1\frac{3}{16}$	3	$14\frac{1}{4}$	12	$\frac{7}{8}$	$3\frac{5}{8}$	2970
12	.79	1470	19	$1\frac{1}{4}$	$3\frac{1}{4}$	$16\frac{1}{2}$	12	$\frac{7}{8}$	$3\frac{3}{4}$	4280
14	.864	1600	21	$1\frac{3}{8}$	$3\frac{1}{2}$	$18\frac{3}{4}$	12	1	$4\frac{1}{4}$	4280
15	.904	1600	$22\frac{1}{4}$	$1\frac{3}{8}$	$3\frac{5}{8}$	20	16	1	$4\frac{1}{4}$	3660
16	.946	1600	$23\frac{1}{2}$	$1\frac{7}{16}$	$3\frac{3}{4}$	$21\frac{1}{4}$	16	1	$4\frac{1}{4}$	4210
18	1.02	1690	25	$1\frac{9}{16}$	$3\frac{1}{2}$	$22\frac{3}{4}$	16	$1\frac{1}{8}$	$4\frac{3}{4}$	4540
20	1.09	1780	$27\frac{1}{2}$	$1\frac{11}{16}$	$3\frac{5}{8}$	25	20	$1\frac{1}{8}$	5	4490
22	1.18	1850	$29\frac{1}{2}$	$1\frac{13}{16}$	$3\frac{5}{8}$	$27\frac{1}{4}$	20	$1\frac{1}{4}$	$5\frac{1}{2}$	4320
24	1.25	1920	$31\frac{1}{2}$ 32	$1\frac{1}{4}$ $1\frac{7}{8}$	$3\frac{3}{4}$ 4	$29\frac{1}{4}$ $29\frac{1}{2}$	20	$1\frac{1}{4}$	$5\frac{1}{2}$	5130
26	1.30	1980	$33\frac{3}{4}$ $34\frac{1}{4}$	$1\frac{7}{8}$ 2	$3\frac{7}{8}$ $4\frac{1}{8}$	$31\frac{1}{4}$ $31\frac{3}{4}$	24	$1\frac{1}{4}$	$5\frac{1}{4}$	5030
28	1.38	2040	36 $36\frac{1}{2}$	$1\frac{15}{16}$ $2\frac{1}{16}$	4 $4\frac{1}{4}$	$33\frac{1}{2}$ 34	28	$1\frac{1}{4}$	6	5000
30	1.48	2000	38 $38\frac{3}{4}$	$1\frac{1}{2}$ $2\frac{1}{8}$	4 $4\frac{3}{8}$	$35\frac{1}{2}$ 36	28	$1\frac{3}{8}$	$6\frac{1}{2}$	4590
36	1.71	1920	$44\frac{1}{2}$ $45\frac{3}{4}$	$1\frac{3}{4}$ $2\frac{3}{8}$	$4\frac{1}{4}$ $4\frac{7}{8}$	42 $42\frac{3}{4}$	32	$1\frac{3}{8}$	$6\frac{1}{2}$	5790
42	1.87	2100	51 $52\frac{3}{4}$	$1\frac{7}{8}$ $2\frac{5}{8}$	$4\frac{1}{2}$ $5\frac{3}{8}$	$48\frac{1}{2}$ $49\frac{1}{2}$	36	$1\frac{1}{2}$	$7\frac{1}{4}$	5700
48	2.17	2130	$57\frac{1}{2}$ $59\frac{1}{2}$	2 $2\frac{3}{4}$	$4\frac{3}{4}$ $5\frac{3}{4}$	$54\frac{3}{4}$ 56	44	$1\frac{1}{2}$	$7\frac{3}{4}$	6090

A thin strip of some specially prepared material is placed between the flanges as shown in Fig. 18. Such a strip is called a **gasket** or **packing**. Sheet rubber, paper, or copper are commonly used though a gasket made of thin sheets of corrugated copper as shown in Fig. 19 gives better service when high pressures are used.

GEARING.

Gearing is the general name for the means of transmitting motion. It includes friction, toothed, belt, rope and wire rope gearing. Wheels which transmit motion from one shaft to another are called gears.

Friction Gearing. In the case of friction gearing, two wheels are pressed together at their circumferences, and if the frictional resistance to slipping is greater than the power transmitted, the driver will cause the follower to rotate. This gearing may be used for transmitting small powers only. To increase the friction the drivers are faced with rubber, leather, wood, or paper. Sometimes grooves are cut in the rims. When wood is used the grain lies along the surface of contact as shown in Fig. 20. With paper or leather the edges of the layers are used as the working surfaces. The followers are usually made of cast iron.

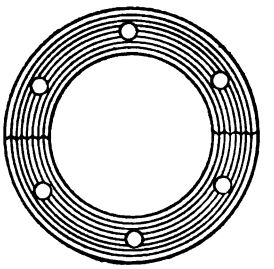


Fig. 19.

To calculate the power transmitted by friction gears:

In Fig. 21 let F = driving force at the circumference of the wheels,

W = force pressing the wheels together,

f = coefficient of friction between the surfaces.

$$\text{Then } F = W f, \text{ or } W = \frac{F}{f}$$

The coefficient of friction, f, has the following approximate values:

Metal on metal	$f = .15$ to $.25$
Wood on metal	$f = .25$ to $.35$
Leather on metal	$f = .25$ to $.30$
Paper on metal	$f = .20$

Let F = The force acting,

r = the radius in inches,

N = the number of revolutions per minute,

H. P. = the horse-power transmitted.

Then, as we have seen in the study of shafts, keys, etc.,

$$F\ r = 63{,}025\ \frac{\text{H. P.}}{N},$$

$$\text{or, H. P.} = \frac{F\ r\ N}{63{,}025}.$$

Suppose a wheel faced with wood drives a cast iron wheel 8 feet in diameter at 200 revolutions per minute. If the force pressing the wheels together is 500 pounds and $f = .3$, what is the H. P. transmitted?

$$F = W f$$

$$F = 500 \times .3 = 150$$

$$r = \frac{3 \times 12}{2} = 18 \text{ inches},$$

$$\text{H. P.} = \frac{F\ r\ N}{63{,}025} = \frac{150 \times 18 \times 200}{63{,}025} = 8.56.$$

The proportions for hubs, arms, etc., are the same as those for spur gearing and will be discussed under toothed gearing.

TOOTH GEARING.

Material. Gear wheels are made of cast iron, cast steel, brass, gun-metal, etc. For large heavy work where accuracy is not essential, cast gears, usually of cast iron, are used. If greater strength is needed, cast steel is used. Gears are cut in blank rims for accurate work. When the linear velocity is great it is usual to make wooden cogs for one wheel of the pair.

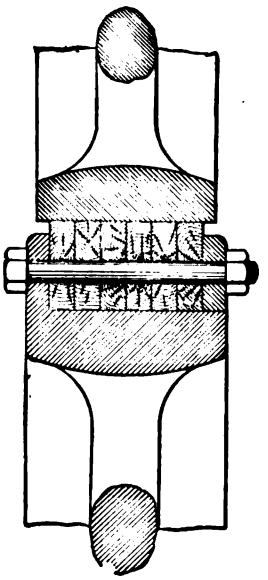


Fig. 20.

In casting gears, formerly the mould was made from a complete wood pattern; but now a method of moulding the entire wheel from a few teeth is used. This method gives greater accuracy because the pattern doesn't become warped. Accuracy is also obtained by cutting the teeth in blank rims by means of a milling machine.

The relative motion of a pair of well made toothed gears is the same as that

of a pair of friction wheels. The surfaces of contact of friction gears are called **pitch surfaces.** Toothed gear wheels are similar to friction gears, and may be considered as short cylinders having projections and indentations at the surface. In making gear teeth the cylindrical surface disappears but the position it occupied is called the pitch surface. In designing, a circle called the **pitch circle** is first drawn; the teeth are then outlined so that the projections or teeth of one wheel fall into the indentations or **spaces** of the other.

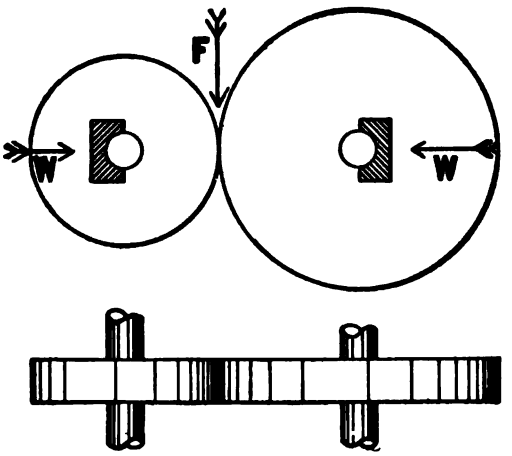


Fig. 21.

It is important that the gears should have a constant velocity ratio, as irregular motion is injurious to the machinery and if there is much backlash the gears run with considerable noise. The width of the space is a little greater than the thickness of the tooth. This small space is called **backlash.** In accurately cut gears the backlash is very small, almost nothing.

The following refers to Fig. 22.

The **circular pitch** is the distance, measured on an arc of the pitch circle, SS, from the face of one tooth to the face of the next; or it is the distance from center to center of the teeth. It is the distance U V.

The **pitch** consists of two parts, the thickness of a tooth and the width of the space between it and the next tooth.

The **depth** of the tooth is the length in the direction of the radius, or X P.

The depth is divided by the pitch circle into two parts; the length of the tooth above the pitch line U P called the **addendum**, and the length below the pitch line U X called the **dedendum**.

The depth of the space is a little greater than that of the tooth, the excess called the **clearance**, being given for inaccuracies of construction. W X, Fig. 22, is the clearance.

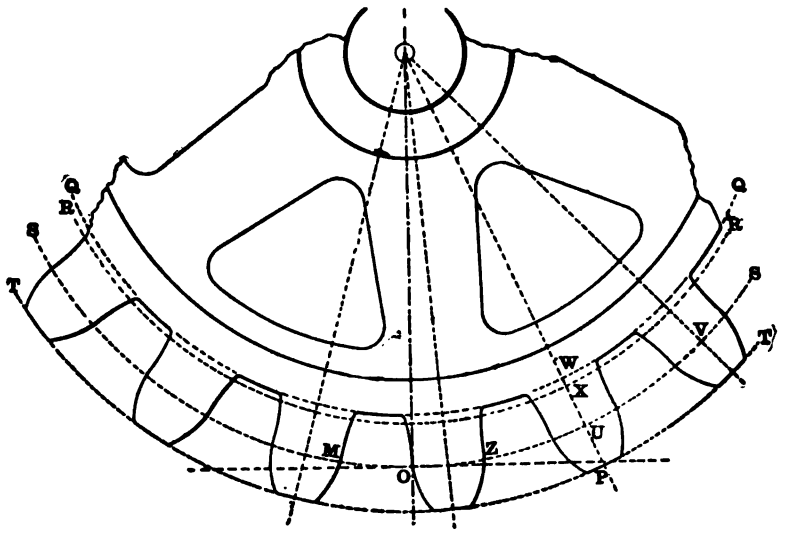


Fig. 22.

In Fig. 22, S S is the pitch circle, Q Q the clearance circle, R R the root or dedendum circle and T T the addendum circle.

The addendum and dedendum are sometimes made equal.

Let P = the pitch,
Addendum = UP = .35 P,
Dedendum = UW = .40 P,
Thickness = OZ = .45 P,
Space = MO = .55 P (about).

In order that the gears may run with constant velocity ratio, little wear and noise, the working faces of the teeth are formed of certain curves. The curves should be such that when two teeth are in contact, the common normal to the tooth curves at the

point of contact passes through the point of contact of the two pitch circles. The curves used are the **cycloid** and **involute.**

If a circle rolls on a straight line the curve traced by a point on the circumference is called a cycloid. In order that the curve may be a perfect cycloid the length of the straight line must be exactly equal to the length of the circumference of the rolling circle; *i. e.* the circle must roll without slipping.

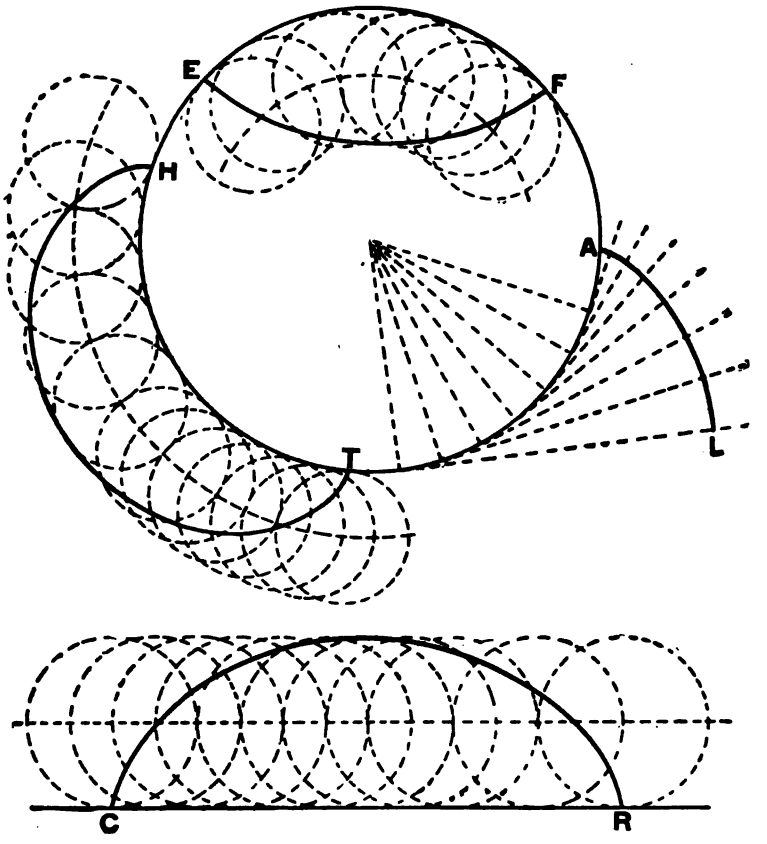


Fig 23.

The curve described by a point on the circumference of a circle that rolls on the outside of another is called an **epicycloid.** If the circle rolls on the inside, the curve is called an **hypocycloid.**

When a thread or string is wound about a circular body and

is then unwound from one end with constant tension, the end describes a curve called the **involute** of the circle. In Fig. 23 the curve C R is a cycloid; the curve H T is an external epicycloid and E F, the internal epicycloid. The curve A L is called the involute.

Circular and diametral pitch. We understand that the pitch surfaces for friction gearing are the cylindrical or conical surfaces in contact; for toothed gears the cylindrical and conical surfaces disappear but their imaginary positions are called the pitch surfaces. The cross-section of the cylindrical surface is a circle and its diameter is called the pitch diameter. In the case of bevel gears having a conical surface, the cross-section is a frustrum of a cone, but as in the case of spur gears there is an imaginary circle called the **pitch circle.** The diameter of the pitch circle is called the pitch diameter, and the distance from the center of one tooth to the center of the next is called the *circular pitch* or simply the *pitch.*

A gear wheel is 22 inches in diameter and the pitch is $1\frac{3}{4}$ inches, how many teeth has it?

The circumference is $\pi\ d$ or $3.1416 \times 22 = 69.115$ inches and the number of teeth $= \frac{69.115}{1.75} = 39.49$. It is evident that a gear must have a whole number of teeth and since the ratio of the circumference of a circle to its diameter is not a whole number, some factor, *i.e.*, diameter, pitch or number of teeth must be a decimal. It has been found from experience that it is most convenient to make the number of teeth and the diameter of the pitch circle whole numbers and the circular pitch a decimal.

Suppose we have a gear wheel eighteen inches in diameter and having 27 teeth. Then the pitch is the circumference of the 18 inch circle divided by 27, or $\frac{56.549}{27} = 2.0944$ inches.

Suppose we divide the *number of teeth* by the diameter of the pitch circle, we get $\frac{27}{18} = 1\frac{1}{2}$, that is, there are $1\frac{1}{2}$ teeth for every inch of diameter. This ratio of the number of teeth to the diameter is called the **diametral pitch,** which for the above gear, is $1\frac{1}{2}$ diametral pitch or simply $1\frac{1}{2}$ pitch.

Another way of finding the diametral pitch, when the circular pitch is known, is from the equation,

$$\text{diametral pitch} = \frac{3.1416}{\text{circular pitch.}}$$

In the above example,

$$\text{diametral pitch} = \frac{3.1416}{2.0944} = 1.5$$

The relations in terms of the diametral pitch are much more simple than those of the circular pitch. A wheel must have a whole number of teeth and it is more convenient if the pitch diameters are whole numbers or whole numbers with easy fractions as $\frac{1}{2}$, $\frac{1}{4}$, and $\frac{1}{8}$. This will be the case if the diametral pitch is used rather than the circular.

Let P = the circular pitch,
D = the diametral pitch,
n = the number of teeth,
d = the diameter of the pitch circle,

Then $d = \frac{n}{D}$,

and, $n = d \times D$.

Therefore, if we have any two of the three quantities, diametral pitch, diameter of pitch, or number of teeth given we can easily find the third.

Suppose we have a gear whose diametral pitch is $1\frac{1}{8}$ and there are 36 teeth. What is the diameter of the pitch circle?

$$d = \frac{n}{D}$$
$$= 36 \div 1\tfrac{1}{8}$$
$$= 32 \text{ inches.}$$

If the diametral pitch is $1\frac{1}{8}$, the circular pitch is,

$$P = \frac{3.1416}{1\frac{1}{8}} = 2.793 \text{ inches.}$$

FOR PRACTICE.

A spur gear has 40 teeth and is 16 inches in diameter. Find the diametral and circular pitches.

Ans. D = $2\frac{1}{2}$, P = 1.257 inches.

The following table shows the relation of diametral pitch to circular pitch. It is very convenient in calculation.

RELATION OF DIAMETRAL TO CIRCULAR PITCH.

Diametral Pitch.	Circular Pitch in inches.	Diametral Pitch.	Circular Pitch in inches.
1	3.142	11	.286
1½	2.094	12	.262
2	1.571	14	.224
2¼	1.396	16	.196
2½	1.257	18	.175
2¾	1.142	20	.157
3	1.047	22	.143
3½	.898	24	.131
4	.785	26	.121
5	.628	28	.112
6	.524	30	.105
7	.449	32	.098
8	.393	36	.087
9	.349	40	.079
10	.314	48	.065

Shrouded Gears. A gear has its teeth shrouded if the rim of the wheel is made wider than the breadth of the teeth and carried outward, uniting the ends of the teeth. Fig. 24 shows three methods of shrouding. At A, the shrouding is carried to the

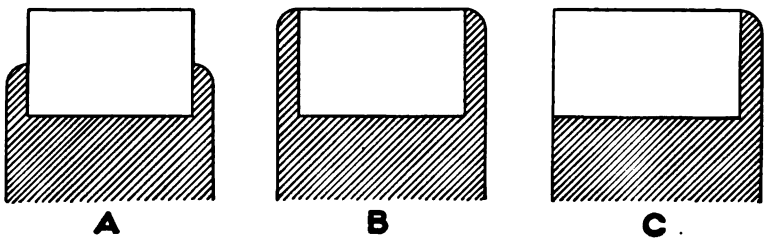


Fig. 24.

pitch line; consequently both wheels may be shrouded. When the construction is as shown at B, only one of the pair can be shrouded, since the shrouding extends to the ends of the teeth on both sides. This method is often used with pinions having few teeth because the teeth of pinions are usually weak at the roots;

also pinions wear faster than larger gears on account of the greater speed. The form shown at C has its shrouding on one side only.

If both wheel and pinion are shrouded (shown at A) the strength of the pair is considerably increased. The difficulty of casting such wheels makes them uncommon.

STRENGTH OF GEAR TEETH.

The strength of gear teeth and the power transmitted depends upon several factors that may vary considerably. Among these are; 1. The strength of the metal; in the case of cast iron, this element is subject to wide variation. 2. The consideration of the position of load, whether on one tooth or distributed among several. 3. The shape of the tooth, that is, the relation of the thickness at the root to the pitch and length. 4. The point at which the load is taken. It may be assumed at the pitch line, at the extreme edge, at an outer corner or spread along the face. 5. The velocity and the factor of safety; the velocity may cause the tooth to break by shock.

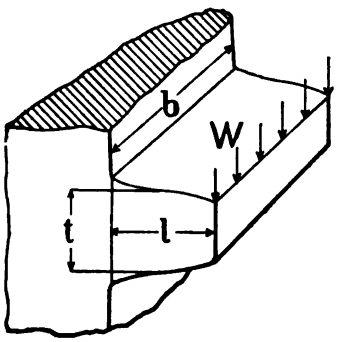


Fig. 25.

In calculating the strength of gear teeth, the curves are not considered. The tooth is taken as a rectangular cantilever whose depth, t, is equal to the thickness of the tooth at the pitch line. The breadth, b, equals the breadth of the tooth, and the length, l, equals the length of the tooth from the root to the end. Fig. 25 shows the above dimensions of the tooth.

The inaccuracies of construction prevent the pressure from being uniformly distributed over the teeth in contact. In order to provide for this, the whole load is considered as being on the edge of one tooth, which makes each tooth a cantilever beam loaded at the end.

From Mechanics the formula for this case is,

$$W = \frac{S}{l} \times \frac{I}{c}.$$

$$\text{in which } I = \frac{b\,t^3}{12}, \text{ and } c = \frac{t}{2}.$$

Then the formula becomes,

$$W = \frac{S}{l} \times \frac{b\,t^2}{6}.$$

The values of S vary with the conditions and are approximately,

9,500 pounds per sq. in. for cast iron, little shock,
6,000 pounds per sq. in. for cast iron, moderate shock,
4,300 pounds per sq. in. for cast iron, great shock,
2,700 pounds per sq. in. for wood.

Let P = the circular pitch. The thickness, t, is usually about $\frac{1}{2}$ P, and l = .7 P to .75 P.

From the study of shafts, keys, etc., we get the formula,

$$F\,r = \frac{63{,}025 \text{ H. P.}}{N}$$

Suppose a gear wheel of cast iron is 28 inches in diameter, makes 60 revolutions per minute, and transmits 10 horse-power. What is the pitch and number of teeth?

$$F\,r = \frac{63{,}025 \text{ H. P.}}{N} \text{ or, } \quad F = \frac{63{,}025 \text{ H. P.}}{r\,N}$$

$$= \frac{63{,}025 \times 10}{14 \times 60} = 750 \text{ (about)}$$

The F of this formula is the same as the W of the equation for cantilever beams. b is usually $2\frac{1}{2}$ times the circular pitch, P. Then as t = .5 P and l = .7 P, we have

$$W = \frac{S}{.7\,P} \times \frac{2.5\,P \times .25\,P^2}{6}$$

Assuming S as 4,000 pounds

$$750 = \frac{4{,}000 \times 2.5\,P \times .25\,P^2}{.7\,P \times 6}$$

$$P = 1.1225$$

If we take P as 1.142 instead of 1.1225 we get the diametral pitch to be $2\frac{3}{4}$ and the number of teeth as $2\frac{3}{4} \times 28 = 77$.

Since P = 1.142, $b = 1.142 \times 2\frac{1}{2} = 2\frac{7}{8}$ inches (nearly) and $l = 1.142 \times .7 = \frac{13}{16}$ (nearly).

If we calculate the circular pitch, we may find from the table the pitch nearest the calculated circular pitch, and from the circular pitch we can find the corresponding diametral pitch. To find the number of teeth we multiply the diametral pitch by the diameter.

Example. A certain spur wheel is to be 24 inches in diameter and is to transmit 15 horse-power while running at 45 revolutions per minute. Find the number of teeth. Assume S = 7,000, $b = 2.5$ P, $t = .5$ P, and $l = .7$ P.

$$F = \frac{63{,}025 \text{ H. P.}}{12\,n} = \frac{63{,}025 \times 15}{12 \times 45} = 1{,}750 \text{ (about)}$$

$$W = \frac{S \times b \times t^2}{l \times 6}$$

$$1{,}750 = \frac{7{,}000 \times 2.5\text{ P} \times .25\text{ P}^2}{.7\text{ P} \times 6}, \text{ from which}$$

$$P^2 = 1.68$$

$$P = 1.29$$

From the table we find 1.257 to be the nearest circular pitch, and the corresponding diametral pitch is $2\frac{1}{2}$. Therefore, the wheel will have $2\frac{1}{2} \times 24 = 60$ teeth.

Durability of Teeth. Teeth should be so designed that the contact will extend over a considerable area of the tooth. They should roll on each other and not slide. Increasing the length, increases the amount of sliding. Usually it is better to make the breadth greater, rather than to increase the length. In ordinary practice the breadth is from two to four times the circular pitch. The following formula is often used to ascertain the durability of the teeth,

$$b = 4\tfrac{1}{2} \times \frac{\text{H. P.}}{\text{D}}$$

in which b = the breadth in inches,
H. P. = the horse-power transmitted,
D = the diameter of the wheel in inches.

In the problem on page 30, we have

$$b = 4\tfrac{1}{2} \times \frac{\text{H. P.}}{\text{D}} = 4\tfrac{1}{2} \times \frac{10}{28} = 1.60$$

and since $b = 2\frac{7}{8}$, the dimension fulfils the conditions.

EXAMPLES FOR PRACTICE.

1. If the pitch is 1.142 inches, what is the addendum, dedendum and thickness?

2. If the diametral pitch is 7, how many teeth in a gear that is 10 inches in diameter?

3. What is the circular pitch of the above gear? (Use table)

4. A gear has 43 teeth and is 12 inches in diameter; what is the length of the addendum?

5. A spur wheel transmits 12 horse-power while running at 60 revolutions per minute. If it is 20 inches in diameter, how many teeth are there on the rim? Assume S = 6,800 pounds.

Ans. 55 teeth.

6. A gear wheel having wood cogs is 24 inches in diameter. It transmits 6 horse-power at 100 revolutions per minute; how many teeth has it? Ans. 84 teeth.

PROPORTIONS OF GEAR WHEELS.

There are several ways to design the various parts of gear wheels. Some parts are designed from calculation; others as

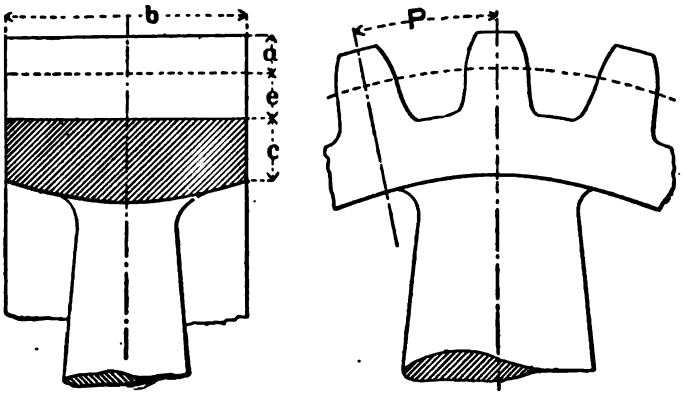


Fig. 26.

some part of the pitch and others from proportions that have given good results.

Rims. The teeth of gear wheels are either cast on the rim or mortised into it. The section of a common form of the rim of a spur wheel is shown in Fig. 26. The rim section is altered

slightly for various forms of arms. The **thickness** c of the rim is made equal to the thickness of a tooth on the pitch circle for large circumferential pitches. If the pitch is small, that is, less than $1\frac{1}{2}$ inches the following formula may be used,

$$c = .4\ P + \frac{1''}{8},$$

which in terms of the diametral pitch is,

$$c = \frac{1.25}{D} + \frac{1''}{8}$$

To find the thickness of the rim of a gear when the circular pitch is 1.047 inches.

$$c = .4\ P + \frac{1''}{8}.$$
$$= .4 \times 1.047 + .125$$
$$= .5438 \text{ inch or } \tfrac{9}{16} \text{ inch.}$$

For a gear wheel having a diametral pitch of 8 the thickness of rim would be,

$$c = \frac{1.25}{D} + \frac{1''}{8}$$
$$= \frac{1.25}{8} + \frac{1''}{8}$$
$$= .281 \text{ inch or } \tfrac{5}{16} \text{ inch.}$$

The **breadth,** b, is 2 or 3 times the pitch. The dimensions a and e are the addendum and dedendum of the tooth; a is made $= .3\ P$ to $.35\ P$ and e, $.35\ P$ to $.4\ P$. The **total depth** of the tooth, the sum of the dimensions a and e, is about $\frac{3}{4}$ the pitch.

Fig. 27 is the section of a rim of a bevel gear. The thickness is a little greater than for spur wheel teeth usually; about $\frac{6}{5}\ c$, where c is found by the formula for spur wheels. The breadth b and the dimensions a and e are the same as for spur wheels.

If wooden teeth, or cogs as they are usually called, are mortised into a gear wheel, the rim thickness, c, is made about $\frac{7}{8}\ P$; for greater strength a and e are less than for cast iron teeth, a being equal to about $.25\ P$ and e, $.3\ P$. The mean thickness i is about $.45\ P$.

Often only one of a pair of wheels is furnished with cogs, the other having iron teeth. In this case the wooden cogs are made about $1\frac{1}{2}$ times the thickness of the iron teeth which mesh with

them. A common method of fastening cogs to the rim is shown in Fig. 28. The cogs may be fixed by round iron pins as shown in Fig. 29.

In Fig. 29 the cogs are fixed in the rim of a bevel gear. The meeting point of the radiating lines is the intersection of the shafts of the bevel wheels. The proportions, in terms of the circular pitch are,

$a = .3\ P$	$e = .35\ P$
$b = 2$ to $3\ P$	$i = .45\ P$
$c = P$	$i' = .35\ P$

Arms. The shape of the cross section of the arms of gears varies with the form of rim. Fig. 30 shows the ordinary forms. The elliptical form shown at A is suitable for light gears. The cross-shaped arm B is generally used for spur gears; the T shaped section shown at C for bevel gears and the I section D for heavy gears.

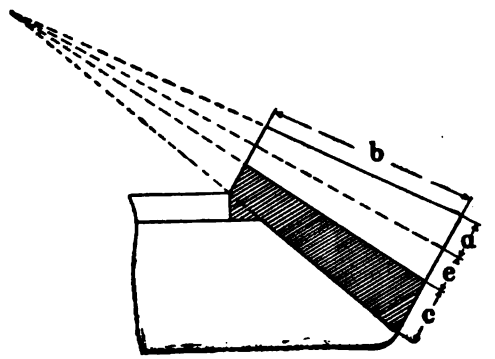


Fig. 27.

The **number of arms** is more a matter of experience than calculation. For a gear less than four feet in diameter there are generally not less than 4 arms; for a gear from four to eight feet in diameter, 6 arms; and for one above eight feet diameter, 8 arms. More arms may be used if desirable.

The following formula may be used, however:

$$n = .55 \sqrt[4]{N^2 P},$$

in which n = the number of arms, N = the number of teeth, and P = the circular pitch. For example, how many should there be in a gear wheel having 64 teeth, diametral pitch 2.

The circular pitch corresponding to diametral pitch is 1.571; Then,

$$n = .55 \sqrt[4]{64^2 \times 1.571}$$

$$= 4.92.$$

Probably 6 arms would be used as it is the next even number. The arms of gear wheels are subjected to bending and are

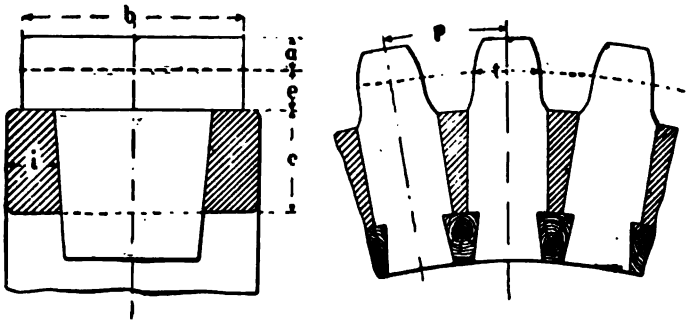


Fig. 28.

considered as fixed at the center and free at the end. If they are cast all in one piece they are fixed at both ends. For safety the first condition is considered. Then let,

n = number of arms,

A = the width of the arm at the center of the wheel,

S = allowable safe stress of the metal,

r = radius of wheel in inches,

p = the pressure between teeth at the pitch circle,

o = the thickness of arm.

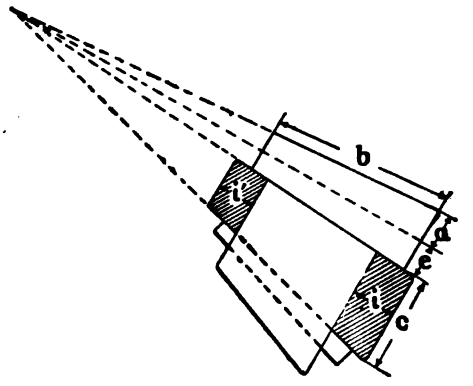


Fig. 29.

In calculating the strength of arms, the ribs are added for lateral stiffness and add very little to the strength.

As castings become weak through strains set up during cool-

ing due to unequal expansion, all parts of the wheel should be of about equal thickness.

Therefore, the thickness of the arm is made about equal to the thickness of the rim which equals the thickness of the tooth. Then $o = c = \frac{1}{2}$ P.

Also on account of the strains set up during cooling, S is given a low value.

Then equating the bending moment $\frac{p\,r}{n}$ to the resistance of the arm $\frac{S\,A^2\,n}{6}$, and inserting the values for o and S, the following formula is obtained,

$$A = \sqrt{\frac{r\,B}{n}}$$

and $o = \frac{1}{3}$ P.

The value of B varies with the width of the face of the wheel,

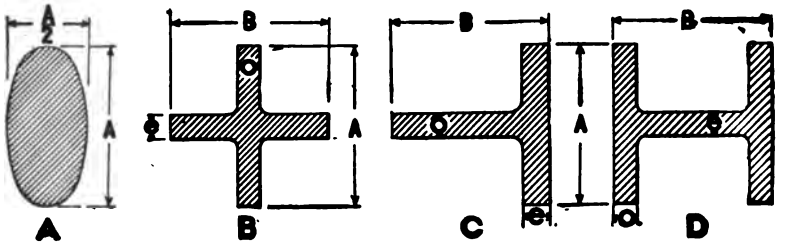


Fig. 30.

from b to $\frac{5}{4}\,b$ at the center and $\frac{3}{4}\,b$ to $\frac{5}{6}\,b$ at the rim. The thickness of the feathers or ribs is usually about .3 P to .4 P and the ribs taper slightly, so that the pattern may be drawn from the mould.

In case the arms have an elliptical cross section as shown at A Fig. 30, the formulas become,

$$A = 1.2\sqrt[3]{\frac{B\,P\,r}{n}}$$

or inserting the value of the diametral pitch in place of the circular pitch,

$$A = 1.75\sqrt[3]{\frac{B\,r}{D\,n}}$$

A gear wheel 25 inches in diameter has 6 arms. If the circular pitch is $1\frac{3}{4}$ and the rim 4 inches wide find the dimensions of the arms.

$$A = \sqrt{\frac{25}{2} \times \frac{4}{6}}$$

$$= 2.88 \text{ or } 2\tfrac{7}{8} \text{ inches.}$$

$$o = \frac{P}{2} = 1\tfrac{3}{4} \div 2$$

$$= \tfrac{7}{8} \text{ inches.}$$

A gear 14 inches in diameter has 4 arms of elliptical cross section. If the diametral pitch is 3 and the width of face $2\frac{1}{2}$ inches, what are the dimensions of the arms?

$$A = 1.75 \sqrt[3]{\frac{2\frac{1}{2} \times 7}{3 \times 4}}$$

$$= 1.98 \text{ inches or } 2 \text{ inches.}$$

The dimensions of the arms produced to the center are found by the above formulas. The arms taper from the center to the rim, the amount being about $\frac{1}{4}$ inch per foot on each side. The thickness o is constant.

In the elliptical form the arm tapers in both width and thickness so that the width is equal to twice the thickness both at the center and at the rim.

Gears may be keyed to the shaft by one key if the gear wheel is small; four keys are often used for large gears.

Hubs or Naves of Gears. In practice, the proportions of the hub of gear wheels depend upon the diameter of the shaft to which the wheel is keyed. The thickness of the hub is often made equal to the radius of the shaft; making the diameter of the hub equal to twice the diameter of the shaft. The length of the nave should be about equal to the width of rim, or b.

In case the shaft is enlarged so as not to be weakened by the cutting of the keyway, the thickness is equal to $\frac{1}{2}$ the diameter of the main portion of the shaft.

Sometimes a wheel transmits only a small portion of the power of the shaft. The above proportions are suitable when the

full power is transmitted. In any case the following formula may be used.

$$h = \tfrac{1}{3}\sqrt[3]{b\ \mathrm{P}\ r}$$

in which h equals the thickness of the hub and the other quantities are the same as previously used.

The hubs of large gears are often split to lessen strains due to unequal cooling, and fastened to the shaft by wrought iron rings shrunk on, as shown in Fig. 31. If h equals the thickness of the hub,

$$a = \tfrac{1}{2}\ h$$
$$b = \tfrac{5}{8}\ h$$
$$c = \tfrac{3}{4}\ h$$

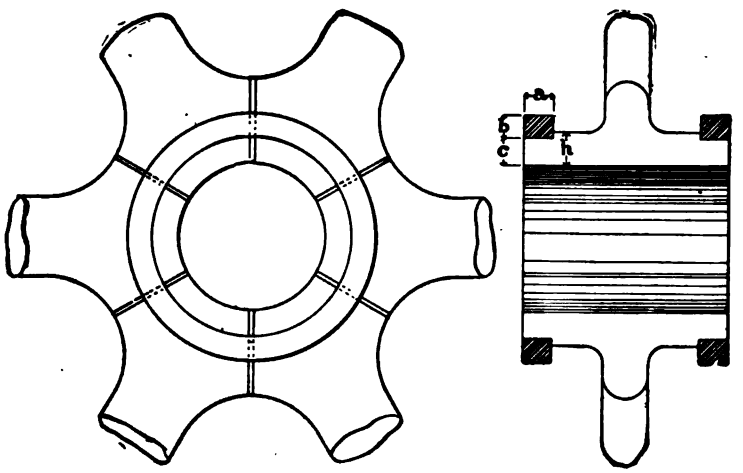


Fig. 31.

Design of a Gear Wheel. A gear transmits 150 horse-power at 80 revolutions per minute. If the pitch circle is 4 feet in diameter and S = 3,750 pounds find the principal dimensions.

$$\mathrm{F} = \frac{63025\ \mathrm{H.\ P.}}{r\ \mathrm{N}}$$
$$= \frac{63025 \times 150}{24 \times 80}$$
$$= 4924$$

$$W = \frac{S}{.7\ P} \times \frac{2.5\ P \times .25\ P^2}{6}$$

$$4924 = \frac{3750}{.7\ P} \times \frac{2.5\ P \times .25\ P^2}{6}$$

$$P = 2.97$$

The nearest pitch in the table to 2.97 is 3.142 which corresponds to 1 diametral pitch. Hence there are $48 \times 1 = 48$ teeth.

Since $P = 3.142$, $b = 3.142 \times 2\frac{1}{2} = 8$ inches.

and $l = 2\frac{3}{16}$ inches.

The thickness of rim could be made equal to the thickness of tooth which would equal about .48 P or,

$$c = .48 \times 3.142$$

$$= 1\tfrac{1}{2} \text{ inches.}$$

The number of arms is found from the equation,

$$n = .55 \sqrt[4]{N^2\ P}$$

$$= .55 \sqrt[4]{48 \times 48 \times 3.142}$$

$$= 5.06.$$

Therefore six arms are used.

The width of arm at the center of the gear.

$$A = \sqrt{\frac{r\ B}{n}} = \sqrt{\frac{24 \times 8}{6}}$$

$$= 5.66 \text{ or } 5\tfrac{5}{8} \text{ inches.}$$

The width of arm at pitch line, allowing a taper of $\frac{1}{4}$ inch per foot for each side is,

$$5\tfrac{5}{8} - 1 = 4\tfrac{5}{8} \text{ inches.}$$

The thickness of the arm is $\frac{1}{2}\ P = 1\frac{5}{8}$ inches.

The thickness of rib is $1\frac{1}{4}$ inches.

The size of shaft is,

$$d = \sqrt[3]{\frac{H}{*n \times .01153}}$$

$$= 5.45 \text{ inches.}$$

A $5\frac{1}{2}$ inch shaft would be chosen.

The thickness of hub,

*n = number of revolutions per minute.

$$h = \tfrac{1}{3}\sqrt[3]{b\,P\,r} = \tfrac{1}{3}\sqrt[3]{8 \times 3.142 \times 24}$$

$$= 2.82 = 2\tfrac{7}{8} \text{ inches.}$$

The length of hub $= b$

$= 8$ inches.

WORM GEARING.

Gearing in which the velocity ratio is not dependent upon the radii of the pitch surfaces is called screw gearing. The most common form of **screw gearing is worm gearing.** The shafts

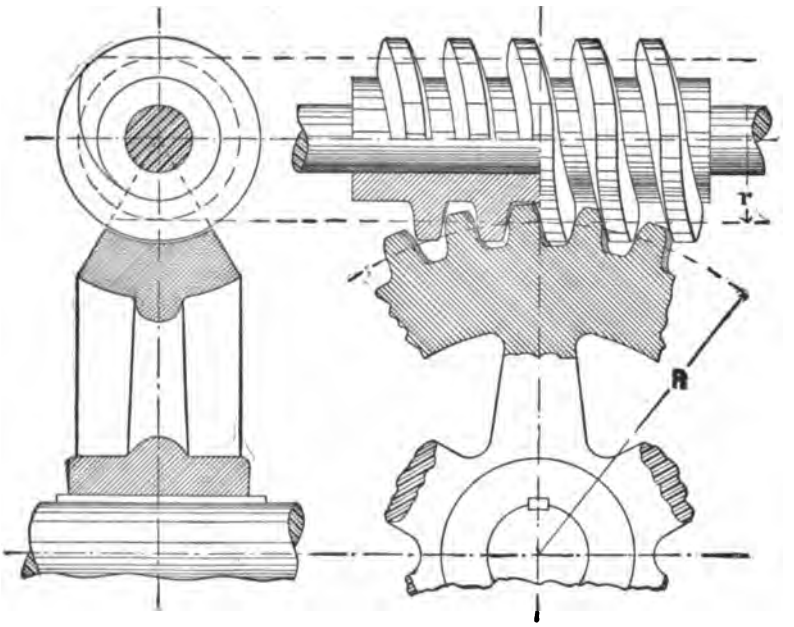


Fig. 32.

are at right angles and a wheel having one thread or sometimes two or three threads meshes with a wheel having many threads. With these wheels a pair of small wheels give a high velocity ratio. The wheel having the few threads is called the worm and the one with the many threads is called the worm wheel.

As the worm is a screw, the worm wheel is rotated a distance equal to the pitch for every revolution of the worm. Thus if the worm wheel has 39 teeth it will rotate $\frac{1}{39}$ of its circumference for

every revolution of the worm. Hence it will take 39 revolutions of the worm to cause one revolution of the worm wheel.

If spur wheels were used to obtain this velocity ratio one of them would have to be many times larger. The worm and worm wheel are usually constructed so that the worm will drive the wheel but the wheel will not drive the worm.

The great disadvantage of this gear is the excessive friction and wear. Worm wheels are similar to spur wheels except that the teeth make an angle with the axis equal to the angle of the threads of the worm.

For accurate work, the teeth of the wheel are cut by a mill ing tool called a *hob*, which is a steel worm made in a lathe. This worm is notched so as to become a milling cutter and then hardened. This gives teeth to the wheel which correspond accurately to those of the worm for that pitch and gives close contact.

Friction of Worm Gearing. One of the greatest disadvantages of worm gearing is the excessive friction caused by the threads of the worm sliding on the teeth of the worm wheel. In designing a worm gear it is, of course, desirable to make the parts of such proportions that the friction may be small.

We can calculate the efficiency by a formula.

For Fig. 32, let R = radius of pitch circle of worm wheel.

r = radius of worm.

p = axial pitch of worm.

E = efficiency of gear.

N = number of teeth in wheel.

n = number of threads in worm.

F = force necessary to turn worm if there were no friction.

P = circular pitch of worm wheel.

f = force acting at pitch line of worm necessary to turn it.

A = the angle between the tangent to worm thread and a plane perpendicular to the axis of worm.

The **efficiency** is the ratio of useful work to total work supplied, or, assuming the coefficient of friction to be about .15,

$$E = \frac{F}{f} = \frac{n\,P}{n\,P + r}$$

It will be seen from the above formula that the smaller the radius of the pitch line of the worm, the greater the efficiency. Also for maximum efficiency the number of threads in worm should be as large as possible.

If the radius of a worm is 3 inches and there are three threads and the circular pitch of the wheel is $\frac{3}{4}$ inch, what is the efficiency?

$$E = \frac{n\,P}{n\,P + r} = \frac{3 \times \frac{3}{4}}{3 \times \frac{3}{4} + 3} = .43 \text{ or } 43\ \%$$

If the radius of the pitch line had been 2 inches the efficiency would be,

$$E = \frac{3 \times \frac{3}{4}}{3 \times \frac{3}{4} + 2} = .53 \text{ or } 53\%$$

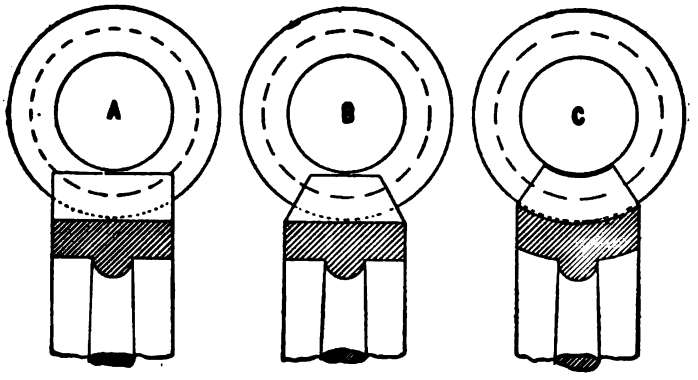


Fig. 33.

A certain worm gear has the following proportions. Pitch of wheel 1.142 inches; the worm has 2 threads and has a radius of $2\frac{1}{4}$ inches. What is the efficiency?

$$E = \frac{n\,P}{n\,P + r} = \frac{2 \times 1.142}{2 \times 1.142 + 2.25} = .50 \text{ or } 50\%.$$

Rims and Arms of Worm Gearing. The rims and arms of worm gearing are designed in the same manner as those for spur gearing having the same diameter and pitch. The width of the face of the worm wheel is generally twice the pitch.

Fig. 33 shows three forms of rims of worm wheel. The one shown at C is the best, since there is line contact between the thread and tooth.

BELT GEARING.

Until about fifty years ago it was thought impossible to satisfactorily transmit considerable amounts of power except by the use of heavy lines of horizontal and upright shafts and trains of gear wheels. Since that time leather belts have been used with belt pulleys and have proved to be economical and efficient.

Material of Belting. Leather belting is commonly made of oxhides tanned by oak bark. The best parts of the hide are cut into strips which are fastened together by lacing, riveting, or cementing.

Cotton belting is made by stitching together from four to ten plies of cotton duck; it is stronger and cheaper, but less durable than leather belting.

Rubber belts are made of several plies of cotton duck cemented together with india rubber. They are well adapted for use in places subjected to extremes of heat, cold, or dampness.

Strength of Belting. Ordinarily, the tensile strength of belting does not enter as a factor in calculations of power transmission. The ultimate tensile strength of leather per inch of width of belt is from 700 to 1,200 pounds. The joints have a strength of only about 200 to 400 pounds per inch of width and the safe working tension is considered to be about one-fourth of these values, 42 pounds per inch of width being a good value to use for single and 80 pounds per inch of width for double belts.

Care of Belts. Leather belts should be well protected from water, steam, or other moisture. Only the finest grained, firmest leather should be used in damp localities. Oil should never be allowed to drop on belts as it destroys the life of the leather. In order to keep a belt in good condition it is well to apply warm tallow when the leather appears very dry. The addition of a little resin to the tallow will preserve the strength of belts used in damp places. There should not be enough resin used to leave the belt sticky and it is bad practice to use a large quantity of

resin upon belts to prevent slipping as such treatment will cause cracking.

Belts should be run with the hair side next the pulley and care taken in twisted belts that the same side of the leather is kept next all the pulleys. The lower side of the belt should be the tight side as with this arrangement the largest possible arc of the pulley is embraced by the belt and hence the driving power of the belt increased. The speed of belting for maximum economy is generally considered to be from 3,500 to 4,500 feet per minute.

Power Transmitted by Belting. (See Mechanics Part I. page 71.)

Belt Pulleys. A flat belt which is placed upon a conical pulley tends to climb toward the larger end. If the pulley is made of a double conical form, that is, having a ridge in the middle, the belt will keep its place and has no tendency to slip off. The rounding of the rim is usually made $\frac{1}{8}$ inch to $\frac{1}{4}$ inch for pulleys up to 12 inches broad and $\frac{1}{8}$ inch to $\frac{1}{4}$ inch per foot of width for wider pulleys. The section of the rim may be constructed by using a radius of from three to five times the width of rim.

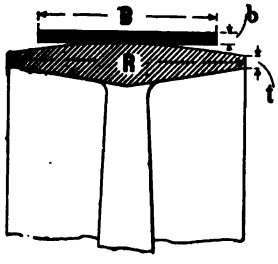


Fig. 34.

Proportions of Pulley. Rim. The rim of a pulley should be made somewhat wider than the belt which it is intended to carry.

The following formula may be used to determine the width of rim R intended to carry a belt of width B.

$$R = \tfrac{9}{8}\,(B + 0.4'')$$

For widths of belts equal to B the following table shows the corresponding values for the width of rim up to 12 inches.

B =	2	3	4	5	6	8	10	12
R =	2.7	3.83	4.95	6.08	7.2	9.45	11.7	13.95
=	$2\frac{3}{4}$	$3\frac{7}{8}$	5	6	$7\frac{1}{4}$	$9\frac{1}{2}$	$11\frac{3}{4}$	14

The general form of rim is shown in Fig. 34.

The thickness of the rim may be calculated from the formula

$$t = 0.7\ b + .005\ D$$

where b is the thickness of belt and D the diameter of pulley.

Arms of the Pulley. The arms of pulleys are generally made of a segmental or elliptical section as indicated in Fig. 35 the elliptical form being preferable.

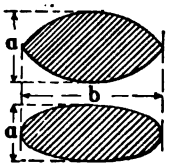


Fig. 35.

The thickness $a = \frac{1}{2} b$ for a segmental arm and for an elliptical arm $a = .4\ b$. Pulleys may have either straight or curved arms. Straight arms are stronger and lighter although curved arms are less liable to fracture from contraction while the casting is cooling.

The section of the arms is smaller near the rim than at the nave. If the breadth and thickness of the arm, produced to the center of the shaft (indicated in Figs. 36, 37 and 38) are represented by b and a, the breadth and thickness at the rim will be $\frac{2}{3} b$ and $\frac{2}{3} a$ respectively.

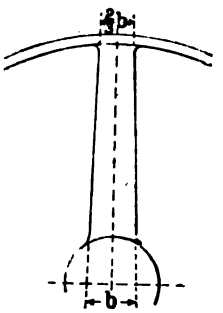


Fig. 36.

In Fig. 36 is shown an ordinary straight arm, in Fig. 37 a curved arm and in Fig. 38 an S-shaped arm. The proportions for the curved arms are clearly indicated. If R is the radius of the pulley measured to the inside of the rim, then in Fig. 37 $d = 0.58$ R and in Fig. 38 $f = 0.47$ R and $h = .24$ R.

Number of Arms. If n is the number of arms, B the

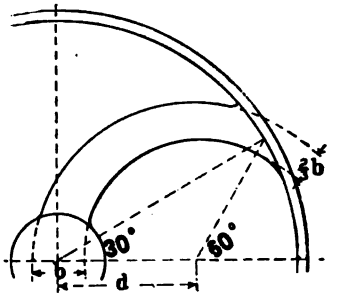


Fig. 37.

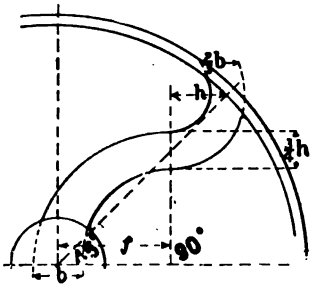


Fig. 38.

breadth and D the diameter of rim the number of arms may be found form the formula,

$$n = 3 + \frac{B\ D}{150}$$

The nearest whole numbers should be taken.

Although the above formula may be used with good results the number of arms may be varied to suit special cases according to the judgment of the designer.

Size of Arms. Let b = breadth of pulley arm, D = diameter of pulley, B = breadth of rim, and n = number of arms, then

$$b = 0.634 \sqrt[3]{\frac{B\ D}{n}} \text{ for single belts,}$$

$$b = 0.798 \sqrt[3]{\frac{B\ D}{n}} \text{ for double belts.}$$

Nave of Pulley. The nave of pulley may have a thickness equal to

$$d = 0.14 \sqrt[3]{B\ D} + \tfrac{1}{4}'' \text{ for single belts}$$

$$\text{and } d = 0.18 \sqrt[3]{B\ D} + \tfrac{1}{4}'' \text{ for double belts.}$$

The length of the nave is generally made equal to about $\frac{2}{3}$ B.

The nave of large pulleys is often strengthened by wrought iron rings shrunk on and if the pulley is to run loose on the shaft it should be bushed with brass.

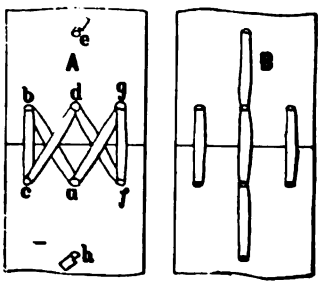


Fig. 39.

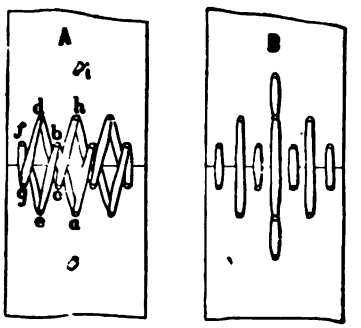


Fig. 40.

Methods for Joining Ends of Belt. The ends of narrow, light belts such as are used on lathes, planers, and other machine tools, are generally fastened together by using strips of white leather tanned with alum. This form of joint is easily made, is flexible, and runs smoothly and noiselessly over the pulleys. Two common methods of lacing are shown in Figs. 39 and 40. To

make a joint, first cut off the ends of the belt squarely. A punch should then be used for making the holes which must not come too near the ends or sides of the belt. Begin to lace in the middle of the belt and lace both sides with equal tightness. In Fig. 39 A is the outside of the belt and B the side running next the pulley. Draw the lacing half way through the middle hole *a* from the under side, then pass it down through *b*, up through *c*, down through *b* and up through *c* again, then down through *d* to *e* where a cut is made upon the end of the lacing which acts as a barb to prevent unlacing.

To lace the other side of the belt, carry the other end up through *d*, down *f*, up *g*, down *f* and up *g* again, down *a*, to *h*.

A somewhat stronger form of lacing is shown in Fig. 40. Bring the lacing up through *a*, and continue through *b*, *c*, *d*, *e*, *f*, *g*, *d*, *e*, *b*, *c*, *h*, and *i*. The edge of the holes should not come nearer than $\frac{7}{8}$ inch from the ends of the belt or $\frac{3}{4}$ inch from the sides and for lacing with two rows of holes, the second row should be at least $1\frac{3}{4}$ inches from the end. For 3-inch belts the number of holes may be the same as in Fig. 39. A 6-inch belt should have seven holes, four in the row nearest the end, and a 10-inch belt, nine holes.

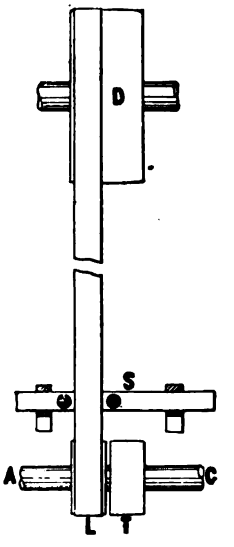


Fig. 41.

For any style of lacing care should always be taken that the lacing on the side of the belt next the pulleys is parallel with the edge of the belt. Joints of heavy belts which transmit large amounts of power are generally made by gluing and riveting.

Belt Guides. Belt shippers are nearly always used where machines derive their power from a line shaft by belts. By using the shipper the machine may be started and stopped without interfering with the motion of the main shaft. Shippers are also used to connect the jack-shaft near the engine with the line shafting. Fig. 41 shows the ordinary arrangement for a shipper. D is a

pulley fast to the driving shaft, S is the shipper and L and T, the loose and tight pulleys on the driven shaft A C.

Where there is considerable tendency for the belt to run off the pulley, the latter is sometimes made with a flange. The flange should, however, be made as shown in Fig. 42 and not as in Fig. 43 because with the latter construction the belt is liable to crowd up against the flange. When made as shown in Fig. 42

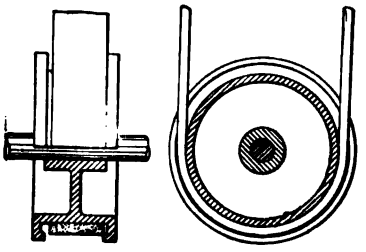


Fig. 42.

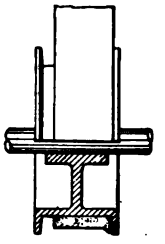


Fig. 43.

the flange will act as a guide to the belt, the latter having little tendency to climb.

BELTS AND PULLEYS TO CONNECT NON-PARALLEL SHAFTS.

Frequently it is required to drive a shaft by another which is at an angle with it. In order to accomplish this the driving pulley must deliver the belt in the plane of the driven pulley; the plane of the driven pulley being one which passes through its middle and at right angles to its axis or shaft.

Quarter-Turn Belt. Fig. 44 shows a quarter-turn belt connecting two pulleys A and B, whose shafts are in parallel planes and at right angles with each other. A is the driver, revolving in the direction indicated, and B is the follower. The driver A, delivers the belt in a plane P through the middle of the pulley B.

If the driver turns in the opposite direction the belt will immediately leave the pulleys because *a*, the point of delivery of the pulley B is not in the plane *b b*, and *c* is not in the plane *d d*.

The angle at which the belt is drawn off the pulleys should not be great because when this is the case the belt is strained, does not hug the pulley well and much power is lost by side slipping. To make this angle as small as possible, the pulleys should be

placed a considerable distance apart, and they should be of small diameter and carry a narrow belt. As wide belts cannot well be used it follows that quarter-turn belts are not suited to transmit large amounts of power.

Suppose we wish to connect the pulley A (Fig. 44) with a pulley B which is the receiving pulley of a machine. If the pulley A revolves in the direction indicated, the middle of the back face of the pulley B should be so located as to just touch a plumb line suspended from the middle of the right hand face of the pulley A. If the pulley A revolves in the opposite direction, the machine together with the pulley B must be moved to the left until a plumb line suspended from the middle of the left hand face of the pulley A will just touch the middle of the back face of the pulley B.

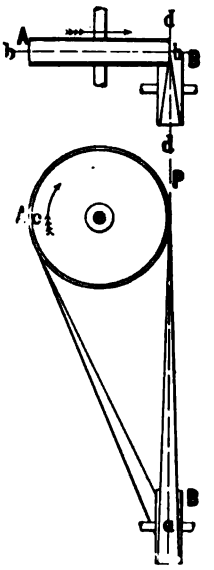


Fig. 44.

A **guide pulley** is often used when two shafts are connected by a quarter-turn belt as indicated in Fig. 45. Here *a b* is the intersection of the two pulley planes of A and B. To locate the plane of the guide pulley G, select a point in the line *a b*, as *c*, and draw lines from this point tangent to the circumferences

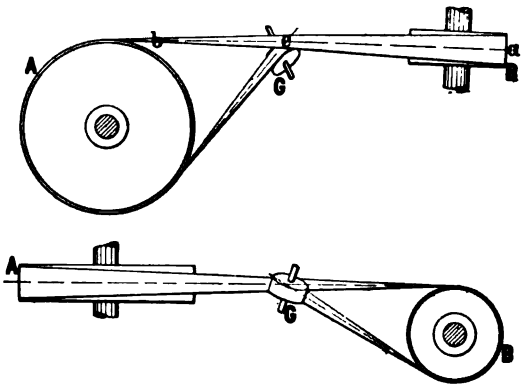


Fig. 45.

of A and B respectively. These lines will determine the plane of the guide pulley. The pulley G is sometimes called a binder pulley

because it tightens the belt. It is often advisable to use one with quarter-turn belts as it causes the belt to hug the pulleys better, thereby increasing the arc of contact.

Fig. 46 and Fig. 47 show other methods of connecting shafts which are at right angles, two guide pulleys being used in each case. The same side of the belt comes in contact with each pulley, except at D (Fig. 47) where it is not possible on account of the crossed belt. The belt can travel in either direction.

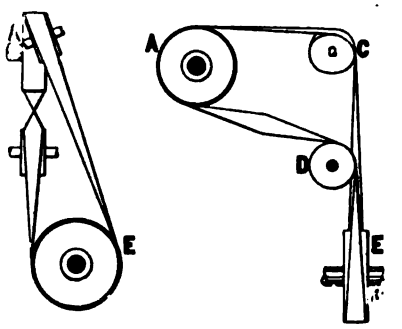


Fig. 46.

When a crossed belt connects two pulleys, as in Fig. 48, it is necessary to give a half twist to the two straight parts of the belt in order that they may pass each other and also to bring the same side of the belt in contact with each pulley.

When two parallel shafts carry two pulleys which are not exactly one above the other, the pulleys may be connected by a belt arranged with two guide pulleys as indicated in Fig. 49. The diameters of the guide pulleys A and B should be equal to the distance between the planes of the pulleys C and D. If the pulleys C and D are not of the same diameter, the arrangement may be made so that a plumb line suspended from the middle of the front face of the pulley A will just touch the middle of the left hand face of the pulley C. Also a plumb line suspended from the middle of the back face of B should touch the middle of the right-hand face of D. In this case the pulleys can only run in the direction indicated by the arrows.

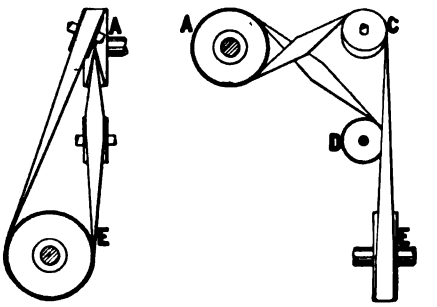


Fig. 47.

Pulleys on shafts which are in the same plane, but at an angle with each other, may be connected as indicated in the right-

hand portion of Fig. 49 the dotted lines showing the arrangement.

The plane of the pulley C in this case, must just touch the circumferences of the pulleys A and B, as shown by the line *cd*.

Pulleys for belts may be arranged in many other ways but the principles which govern the arrangement in each case, are the same. Whenever practicable, belts should be arranged to travel in either direction because engines sometimes move backward a part of a turn. Thus any belt not admitting of motion in either direction, is likely to be thrown off.

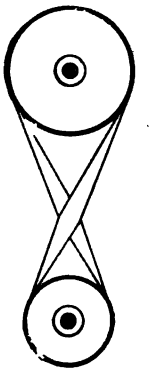


Fig. 48.

Speed Cones and Stepped Pulleys. Speed cones are contrivances for varying the velocity of a shaft by means of a belt. They are usually made of cast iron or wood, their general appearance being shown in Fig. 50. The form shown in Fig. 51 is known as the conoid which is suitable for open belts, whereas the ordinary cone pulley is more suitable for crossed belts.

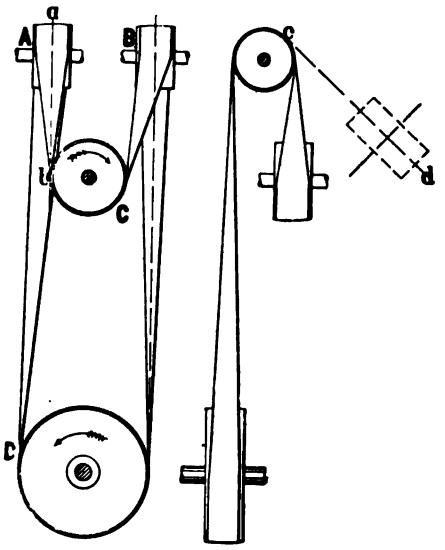


Fig. 49.

Fig. 52 is a cut of the ordinary stepped pulley widely used in machine shops. When designing a pulley of this kind a trapezoid A B C D is first laid out and then a sufficient number of diameters are drawn at equal distances apart upon which to construct the steps.

ROPE GEARING.

The advantages of rope gearing over belt gearing have resulted in the substitution of the former for belt gearing in many factories and mills. The installation of rope gearing is cheaper; there is less loss of power by slipping; power can be transmitted in any direction easier; and there is also the advantage that in case

one of the ropes on the fly-wheel breaks, the work in the entire plant is not interrupted, whereas, if the fly-wheel carries one or two large belts and one of these breaks, a large portion of the work in the plant must stop.

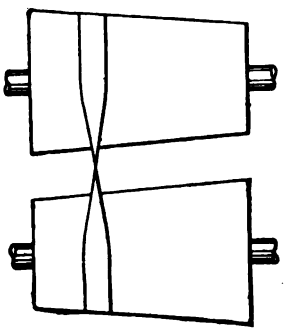


Fig. 50.

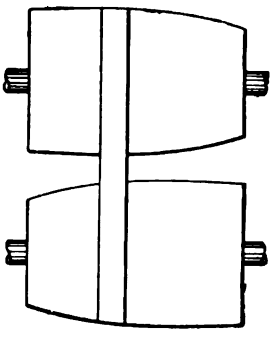


Fig. 51.

As compared with the old system of using gears, the power transmitted by ropes is said to give a much more uniform motion to machinery but this is probably caused by the combination of heavy fly-wheels and high velocities.

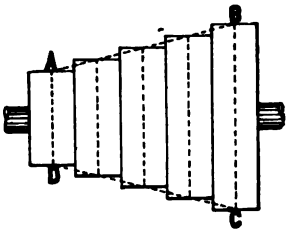


Fig. 52.

Material of Ropes. The ropes are from $\frac{3}{4}$ inch to $2\frac{1}{2}$ inches in diameter. They are usually made of hemp, manilla or cotton, and sometimes round leather ropes are used. Cotton rope is not nearly as strong as hemp rope but is much more durable when the diameter of the pulleys over which it runs is small. Ropes of rawhide are excellent in places where there is much dampness as they are little affected by it.

Strength of Ropes. The strength of cotton transmission ropes may be calculated from the formula $S = 4{,}600\,d^2$, where S is the ultimate breaking strength and d the diameter in inches. By use of the above formula the following table of values has been computed.

STRENGTH OF COTTON TRANSMISSION-ROPES.

Diameter of Rope in Inches. d	Ultimate Breaking Strength. $S = 4{,}600\ d^2$
$\frac{1}{2}$	1,150
$\frac{5}{8}$	1,800
$\frac{3}{4}$	2,600
$\frac{7}{8}$	3,500
1	4,600
$1\frac{1}{4}$	7,200
$1\frac{1}{2}$	10,400
$1\frac{3}{4}$	14,000
2	18,400

The strength of good manilla rope varies from 10,000 pounds per square inch for a 2-inch rope, to about 12,000 pounds for a half-inch rope. In the formula $S = 100\ d^2x$ allowance is made for this fact. S is the breaking strength of the rope, d its diameter, and x a variable depending on the diameter of the rope. The value of x is equal to $81 - 9d$, where d is the diameter in inches. The breaking strength per square inch of actual section will be equal to

$$\frac{S}{0.8\,\frac{\pi\,d^2}{4}}$$

The following table is computed by use of the above formulas

STRENGTH OF MANILLA TRANSMISSION-ROPES.

Dia. of Rope = d.	Breaking Strength. $S = 100\ d^2x$	Strength per sq. in. $\frac{S}{0.8\,\frac{\pi\,d^2}{4}}$
$\frac{1}{2}$	1,900	12,200
$\frac{5}{8}$	2,900	11,950
$\frac{3}{4}$	4,100	11,750
$\frac{7}{8}$	5,500	11,525
1	7,100	11,350
$1\frac{1}{8}$	8,800	11,125
$1\frac{1}{4}$	10,900	10,975
$1\frac{1}{2}$	15,000	10,625
$1\frac{3}{4}$	19,800	10,300
2	25,100	10,000

The **wear** of ropes is both *internal* and *external*. The movement of the fibres upon each other, when under pressure in bending over the sheaves, causes the internal, while the slipping and wedging of the rope in the grooves of the pulley cause the external wear. Both of these classes of wear are proportional to the speed, that is, the greater the speed the greater will be the wear.

In the accompanying diagram (Fig. 53) horizontal distances represent velocities and vertical distances represent horse-powers transmitted by 1, $1\frac{1}{4}$, $1\frac{1}{2}$, $1\frac{3}{4}$, and 2 inch manilla ropes. A is the curve for a 2-inch rope; B, for a $1\frac{3}{4}$-inch rope; C for a $1\frac{1}{2}$-inch rope; D for a $1\frac{1}{4}$-inch rope, and E for a 1-inch rope.

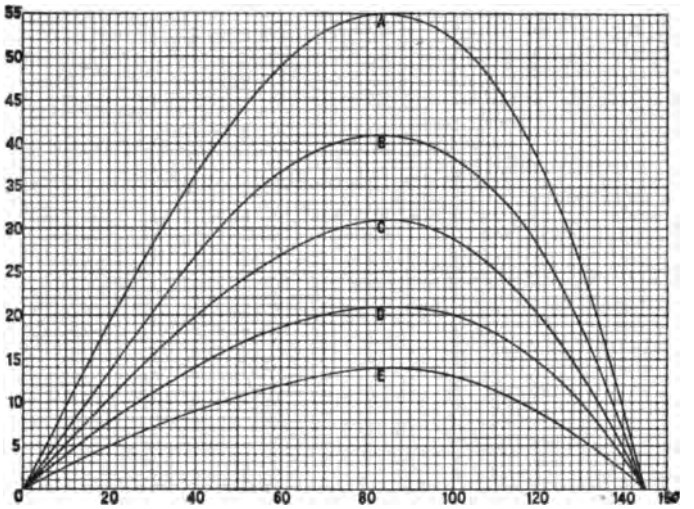


Fig. 53.

It will be seen from the diagram that the greater the velocity the greater the horse-power transmitted up to about 85 feet per second. When a greater velocity than this is attained, however, the centrifugal force partially overcomes the friction between the rope and pulley and, at a speed of about 145 feet per second, the friction is entirely overcome by the centrifugal force and no power is transmitted. From a consideration of the effects of centrifugal force it is generally thought to be inadvisable to run ropes faster than 5000 feet per minute and when the matter of durability is

also taken into account the rope should not be run faster than 3,500 to 4,000 feet per minute.

To obtain the working strength of rope from the breaking strength a factor of safety of 10 should be used. This factor will be large enough to take care of reduction of strength due to the splice, lubrication, imperfection in manufacture and unknown stresses which may be set up in the rope.

Lubrication of Ropes. The fraying on the surface of ropes and the interior wear are somewhat reduced by the use of lubricants. Very often the rope is laid up dry and the exterior is given a coating of dressing which should penetrate to the interior and lubricate the fibres. With some lubricants this may occur but with others the interior is not affected and remains dry. In the latter case, if there is much bending, the rope soon wears out.

The interior wear of cotton ropes being small, internal lubrication is undesirable, but the use of a *surface* lubricant will prevent the surface from fraying and also protects the rope from moisture.

Beeswax and black lead with a little tallow make an excellent moisture proof covering for ropes. It fills in the spaces between the strands and causes the rope to assume a round and smooth appearance like a rod of iron. This coating does not penetrate to the interior and is only suitable for cotton ropes, or where internal lubrication is not needed.

Lard, tallow and other greases are used as lubricants for the interior of ropes and give good results. When a rope is run out of doors and it is desired to protect it from the weather and also to lubricate its interior a mixture of black lead and tallow will be found very satisfactory.

Pulleys for Rope Belting. The pulleys for rope belting are usually of cast iron and have grooves turned in the rim. The larger the pulley the less will be the injury done to the rope in bending and unbending. A pulley should not be less than thirty times the diameter of the rope which runs upon it.

The form of the groove is a very important matter as the durability and transmitting power of the rope are to a great extent dependent upon it. The sides of the groove are usually inclined at an angle of 45° and are generally formed by arcs of circles and

straight lines. In Fig. 54 the diameter of the rope is taken as the unit for constructing the rim-section.

$$a = 1.6\,d \qquad e = \frac{3}{4}d + \frac{1}{16}''$$

$$b = \frac{d}{4} + \frac{3}{16}'' \qquad l = \frac{d}{2}$$

$$c = \frac{d}{4} + \frac{1}{16}'' \qquad i = \frac{d}{3}$$

In large wheels, where the weight of a flat rim would be greater than required for strength, the inside is hollowed out as shown by the dotted line.

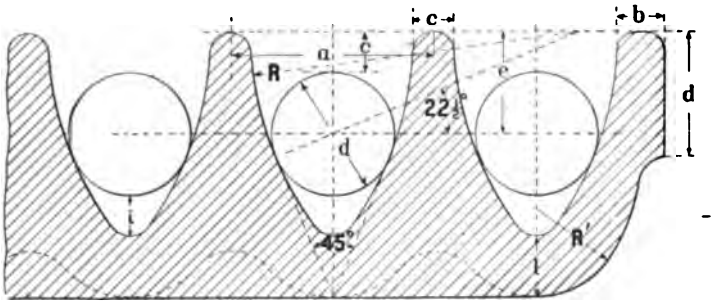


Fig. 54.

The rope does not run upon the bottom of the groove but is wedged between the sides as is shown in the cut.

With long drives it is often necessary to use guide pulleys to support the ropes. The rope rests upon the bottom of the groove of the guide pulley and is not wedged in.

When the rope stretches with use, the extra length is taken up by cutting a piece out of it and resplicing, or else a tension pulley similar to that shown in Fig. 55 is used. The tension pulley may also be used where it is desired to increase the bite of the rope on the pulleys. Owing to the several wraps of the rope around the pulley, the arc of contact is increased and hence a larger amount of power may be transmitted.

The ordinary arrangement of ropes with respect to the flywheel of the engine and rope tower is shown in Fig. 56. The ropes which connect the flywheel with the smaller pulleys hang in

catenary curves and the weight of the rope rather than its tightness is what produces the bite of the rope on the pulleys.

Wire Ropes are used to a considerable extent for long-distance or telo-dynamic transmission of power.

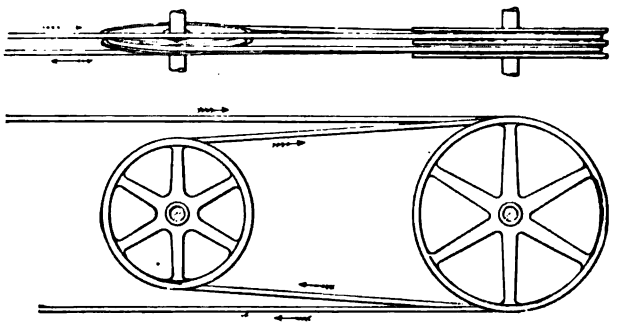


Fig. 55.

The amount of power transmitted by leather belts may be increased in three ways; by increasing the strength of the belt, by

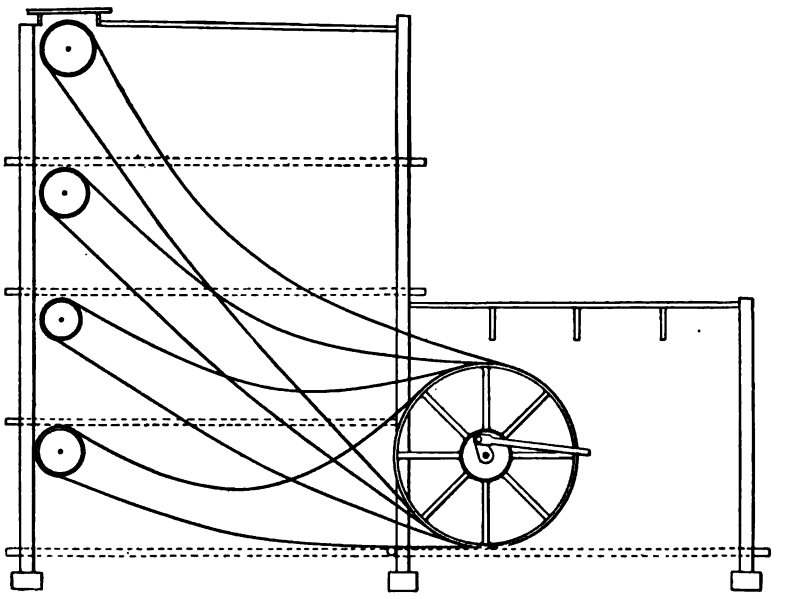


Fig. 56.

increasing the velocity of the belt, or by increasing the frictional bite of the pulleys. This last principle is made use of in ordinary

rope transmission as the rope becomes wedged in the V-shaped grooves in the rim of the pulleys but wire rope cannot be used in V-shaped grooves because of the injury done to the rope.

The principle of telo-dynamic transmission is to use as flexible ropes as possible upon pulleys of large diameter to reduce the loss of work in journal friction. If the power is to be transmitted a great distance there are several relay stations. At each relay station is a horizontal shaft upon which is fixed a double grooved pulley. In one groove of the pulley runs the wire-rope belt from the preceding station which drives the pulley and in the other groove runs the rope which transmits the power to the next station.

The length of each relay is usually from 400 to 500 feet. Power is often transmitted in this way a distance of several thousand yards. If the distance is less than 130 feet, this method of transmission is not suitable owing to the oscillation of the rope which causes loss of power.

The allowable working stress of iron wire may be taken at 25,000 pounds per square inch and of steel wire at 27,000 pounds per square inch. If d is the diameter of each wire of the rope and n the number of wires, the area of cross-section is

$$\frac{\pi d^2}{4} n.$$

This value for the area of cross-section should be used in calculations of tensile strength. When the rope is not bent round a drum or sheave but is simply subjected to a straight pull the area of section $\frac{\pi d^2}{4} n$ should be equal to the total load on the rope divided by the safe stress per square inch, that is,

$$\frac{\pi d^2}{4} n = \frac{\text{total load on rope}}{\text{safe stress per square inch}}.$$

The most pliable variety of iron rope (generally used for hoisting) contains nineteen wires in the strand, there being six strands. Data upon this kind of wire rope is given in the following table:

Diameter in inches.	Circumference in inches.	Weight per foot in pounds. Rope with hemp center.	Breaking strain, tons of 2,000 lbs.	Proper working load in tons of 2,000 lbs.	Circumference of new manilla rope of equal strength.	Min. Size of Drum or sheave in feet.
			IRON.			
2¼	6¾	8.00	74	15	14	13
2	6	6.30	65	13	13	12
1¾	5½	5.25	54	11	12	10
1⅝	5	4.10	44	9	11	8½
1½	4¾	3.65	39	8	10	7½
1⅜	4⅜	3.00	33	6½	9½	7
1¼	4	2.50	27	5½	8½	6½
1⅛	3½	2.00	20	4	7½	6
1	3⅛	1.58	16	3	6½	5¼
⅞	2¾	1.20	11.50	2½	5½	4½
¾	2¼	0.88	8.64	1¾	4¾	4
⅝	2	0.60	5.13	1¼	3¾	3½
9/16	1⅝	0.44	4.27	¾	3½	2¾
½	1½	0.35	3.48	½	3	2¼
7/16	1⅜	0.29	3.00	¼	2¾	2
⅜	1¼	0.26	2.50	¼	2½	1¼
			STEEL.			
2¼	6¾	8.00	155	31		8½
2	6	6.30	125	25		8
1¾	5½	5.25	106	21		7¼
1⅝	5	4.10	86	17	15	6¼
1½	4¾	3.65	77	15	14	5¾
1⅜	4⅜	3.00	63	12	13	5½
1¼	4	2.50	52	10	12	5
1⅛	3½	2.00	42	8	11	4½
1	3⅛	1.58	33	6	9½	4
⅞	2¾	1.20	25	5	8½	3½
¾	2¼	0.88	18	3½	7	3
⅝	2	0.60	12	2½	5¾	2¼
9/16	1⅝	0.44	9	1½	5	1¾
½	1½	0.35	7	1	4½	1½
7/16	1⅜	0.29	5½	¾	3¾	1¼
⅜	1¼	0.26	4½	½	3½	1

It is good practice to allow one fifth to one seventh of the ultimate strength for the safe working load, the value chosen depending somewhat upon the speed so as to get good wear from the rope. As wear increases with the speed it is better to increase the load than the speed. When wire rope is to be substituted for hemp rope it is well to allow for the former the same weight per foot which has been found to work well for the latter. The greater the diameter of the pulleys or drums upon which wire rope is run, the longer the rope will last. The minimum diameter of drum is given in the table.

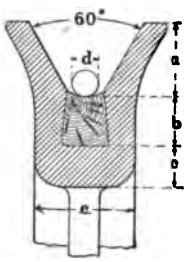


Fig. 57.

Wire rope is manufactured either with a hemp or wire center. The former is the more pliable and is more suitable for use where there is short bending.

Wire rope may be preserved by applying raw linseed-oil with a piece of wool.

Pulleys for Wire Ropes. Wire ropes are injured by the lateral crushing if run upon pulleys with V-shaped grooves. Hence, pulleys are constructed with wide grooves so that the rope may rest upon the bottom. The frictional resistance between the rope and pulley is much increased by lining the bottom of the groove with gutta percha, tarred hemp, rubber, leather or wood. The material used is hammered into the groove which is dovetailed in section, as shown in Fig. 57.

The proportions for the construction of the rim of pulley for wire ropes are shown in Fig. 57.

The unit for the construction of the rim section is d, the diameter of rope.

$$a = 3 \text{ to } 3\tfrac{1}{2}\, d$$
$$b = d + \tfrac{1}{8}'' \text{ to } d + \tfrac{1}{2}''$$
$$c = \tfrac{1}{2}\, d + \tfrac{1}{4}'' \text{ to } \tfrac{1}{2}\, d + \tfrac{3}{4}''$$
$$e = 3\tfrac{1}{2}\, d \text{ to } 4\, d.$$

CHAIN GEARING.

When a considerable amount of power is to be transmitted between two shafts running at a slow speed, metal chains called gearing chains are used with toothed pulleys. The chain belt can-

not slip upon the toothed wheel, and as the belt can be designed with almost any strength an extremely great power may be transmitted. The chief objection to the use of chain gearing is that the chain will in time become stretched and as it then has a greater pitch than the toothed wheel they will work very badly. The acting surface of the pulley must be adapted to the chain so as to insure sufficient hold between them. *Chain drums* have one end of the chain made fast to them, the chain being guided by a suitably formed groove. They are used in cranes.

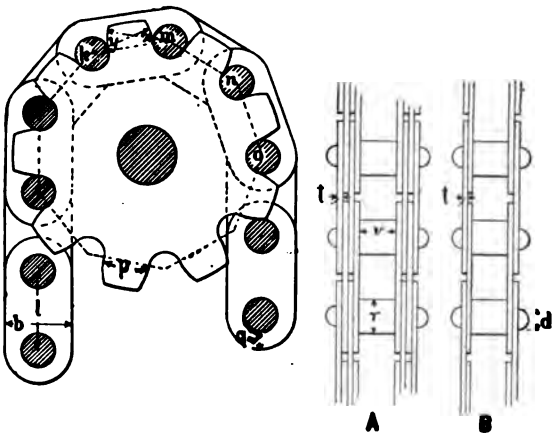


Fig. 58.

Gearing chains are usually made of flat links riveted together at their ends to allow a free turning of the links at the joints. A pulley for such a chain is shown in Fig. 58. It has a polygonal prism, *m n o*, etc., for its pitch surface, and as the links are of the same length, the pitch line is a regular polygon, each side being equal to the distance from center to center of the rivets.

The acting parts of the teeth outside of the pitch line are formed by arcs of circles whose centers are the adjacent vertices of the pitch polygon; thus the arc x is drawn with k as a center, and the arc y with m as a center. The length of the teeth should be such that they project nearly to the outside of the chain when it is in position on the wheel.

The ordinary form of gearing is a flat-linked chain made up of two systems of links separated by an enlargement of the pins

connecting the links as shown in the cut, A being a somewhat stronger chain than B.

Good proportions for the links are as follows, the diameter, d, of the pin in the link eye being taken as the unit.

Length of link, center to center of pins	$l = 2.9\,d$
Breadth of link	$b = 2.5\,d$
Diameter of middle part of pin	$r = 1.2\,d$
Length of middle part of pin	$v = 1.7\,d + 0.2''$
Distance between eye and end of link	$q = 0.85\,d$

Below is given a table using these proportions for chains of 2, 4, 6 and 8 links. The dimensions are in inches.

Load on Chain lbs.	No. of Links	Link Length l	Link Breadth b	Link Thickness t	Pin Diameter at Ends d	Pin Diameter at Center r	Length of Middle Part of Pin v	Width of Tooth $p = l - r = 1.7\,d$
220	2	.55	.39	0.060	0.16	0.20	0.43	0.35
550	2	.83	.65	0.098	0.26	0.32	0.67	0.51
1,100	2	1.10	.93	0.118	0.37	0.47	0.83	0.65
1,650	2	1.42	1.12	0.158	0.45	0.55	0.94	0.75
2,200	4	1.42	1.24	0.158	0.49	0.59	1.02	0.83
3,300	4	1.77	1.52	0.177	0.61	0.75	1.22	1.04
4,400	4	2.01	1.79	0.216	0.69	0.83	1.38	1.18
5,500	4	2.44	2.11	0.256	0.85	1.02	1.61	1.42
8,820	6	2.44	2.11	0.177	0.85	1.02	1.61	1.42
12,100	6	2.76	2.36	0.197	0.94	1.14	1.81	1.61
16,500	6	3.15	2.70	0.216	1.08	1.30	2.05	1.85
22,050	6	3.66	3.15	0.256	1.26	1.54	2.32	2.14
33,070	8	4.17	3.60	0.197	1.44	1.73	2.60	2.44
44,100	8	4.80	4.13	0.236	1.65	1.97	3.00	2.72
55,100	8	5.35	4.61	0.256	1.85	2.24	3.31	3.13
66,100	8	5.87	5.05	0.295	2.03	2.44	3.62	3.43

Crane Chains. Ordinary chains used as simple fastenings are forged from round iron bars of first quality. The greatest stress in each link is at the end and hence, theoretically, the link section should be made somewhat greater at the ends. Links thus constructed would entail much extra expense and as there is, in any case, much uncertainty regarding the strength of the weld, the

section is usually made uniform. The strength of weld averages about 80 per cent. of the strength of the solid bar. When designing a chain the links should be as small as possible because this will result in greater strength and flexibility.

The inside radius at the ends of the links should be a little greater than the radius of the bar of which the chain is made. The chain shown in Fig. 59 is called **close-link** chain and is preferable to **studded chain** shown in Fig. 60. If d is the diameter of the iron, the ordinary proportions for close-link chain are, for the inside diameters 1.5 d and 4 d, and for outside diameters 3.5 d and 6 d. The proportions for studded chain vary greatly, the following being proportions for one style.

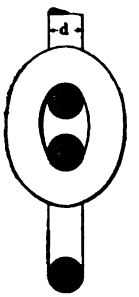


Fig. 59.

Length of link, inside $3d$ to $5d$.
Length of link, outside $5d$ to $7d$.
Width of link, inside $1\frac{1}{2}d$ to $1\frac{3}{4}d$.
Width of link, outside $3\frac{1}{2}d$ to $3\frac{3}{4}d$.
Stud, diameter at ends $0.7d$ to d.
Stud, diameter at center $0.6d$ to $0.8d$.

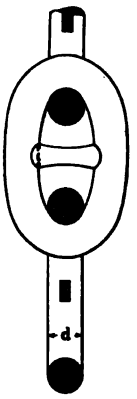


Fig. 60.

The stud renders the chain less liable to kink and also resists the tendency of the link to collapse.

Chain cable is made in lengths of from 10 to 25 fathoms, these lengths being joined by swivels.

Strength of Chains. The strength of chain up to $\frac{1}{2}$ inch diameter is about 23 to 25 tons per square inch, and for chains of 2 inches diameter or more, 18 to 21 tons per square inch. A suitable deduction should be made from these values to allow for the weakening of the chain due to the weld and to bending action.

By the following table the safe working load for chain may be calculated.

Dia. of iron in ins.	$\frac{3}{16}$,	$\frac{1}{4}$,	$\frac{9}{32}$,	$\frac{5}{16}$,	$\frac{3}{8}$,	$\frac{7}{16}$,	$\frac{1}{2}$,	$\frac{9}{16}$,	$\frac{5}{8}$,	$\frac{11}{16}$,	$\frac{13}{16}$
Safe load in tons	.06,	.25,	.5,	.75,	1,	$1\frac{1}{2}$,	2,	$2\frac{1}{2}$,	3,	4,	5

EXAMPLE FOR PRACTICE.

The diameter of iron used in a chain cable is $\frac{1}{4}$ inch. What is the safe load in pounds which it is capable of supporting?

The following table gives the strength and weight of close-link crane chains and the size of hemp cable of equal strength.

Diameter of Iron d in Inches.	Weight of Chain per Fathom.	Breaking Strength in Tons.	Girth of Rope of Equal Strength.	Weight of Rope in lbs. per Fathom.
$\frac{1}{4}$	3.5	1.9	2	$1\frac{3}{8}$
$\frac{5}{16}$	6.0	3.0	$2\frac{1}{2}$	$1\frac{1}{2}$
$\frac{3}{8}$	8.5	4.3	$3\frac{1}{4}$	$2\frac{1}{2}$
$\frac{7}{16}$	11.0	5.9	4	$3\frac{3}{4}$
$\frac{1}{2}$	14.0	7.7	$4\frac{3}{4}$	5
$\frac{9}{16}$	18.0	9.7	$5\frac{1}{2}$	7
$\frac{5}{8}$	24.0	12.0	$6\frac{1}{4}$	$8\frac{1}{2}$
$\frac{11}{16}$	28.0	14.6	7	$10\frac{1}{2}$
$\frac{3}{4}$	31.5	17.3	$7\frac{1}{2}$	12
$\frac{13}{16}$	37.0	20.4	$8\frac{1}{4}$	15
$\frac{7}{8}$	44.0	23.1	9	$17\frac{1}{2}$
$\frac{15}{16}$	50.0	26.1	$9\frac{1}{2}$	$19\frac{1}{2}$
1	56.0	29.3	10	22
$1\frac{1}{8}$	71.0	36.3	$11\frac{1}{4}$	$27\frac{3}{4}$
$1\frac{1}{4}$	87.5	44.1	$12\frac{1}{2}$	$34\frac{1}{2}$
$1\frac{3}{8}$	105.8	52.8	$13\frac{3}{4}$	$41\frac{1}{2}$
$1\frac{1}{2}$	126.0	62.3	15	$49\frac{1}{2}$

Crane Hook. The ordinary form of crane hook is shown in Fig. 61 the diameter d of the iron being taken as the unit of proportion.

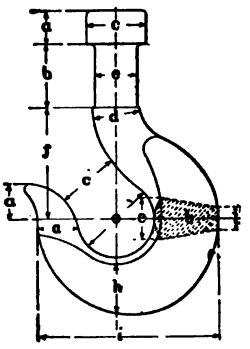


Fig. 61.

$a = 0.75d$ to d $\quad f = 2.25d$ to $2.5d$

$b = 1.4d$ $\quad h = 1.15d$

$c = 1.25d$ $\quad i = 3.75d$

$e = 0.875d$ $\quad l = .25d$

$o = 1.65d$ to $1.75d$

A crane hook is usually required to receive a rope sling and the opening c should therefore be large enough to receive the rope. The section of the hook should be deepest in those parts which are subjected to the greatest bending stress.

and for this reason the hook is flattened as shown in the cut. The inside should be well rounded in order not to injure the rope.

RATCHET WHEELS.

A wheel provided with suitably shaped teeth, which receives an intermittent circular motion from a vibrating piece, is called a *ratchet wheel.*

The rachet wheel in Fig. 62 is represented by A; B is the vibrating piece carrying at one end the *detent* or *catch* C, its other end being pivoted upon the same axis with the ratchet wheel. When the arm B moves left handed the wheel will be pushed for-

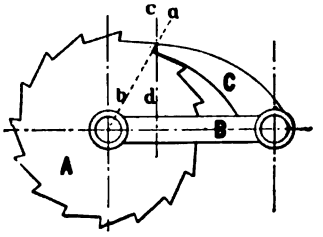


Fig. 62.

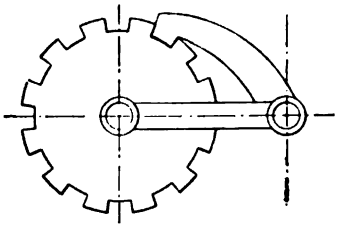


Fig. 63.

ward a certain distance and as B moves back again for the next stroke, the catch will slide over the points of the teeth.

The motion of the arm B must be at least sufficient to cause the wheel to advance one tooth. In order that the catch may retain its hold upon the teeth, a line *a b* drawn through the acting surfaces of the catch and tooth, should pass either through the center of the wheel or to the left of the center of the wheel, the latter construction being preferable. If the line *c d* is the one which passes through the acting surfaces of the tooth and catch, the catch will be thrown outward and will refuse to act. The *effective stroke* of a catch is the space through which the ratchet wheel turns for each forward stroke of the arm.

This form of catch is often used upon ratchet drills such as are used to drill metals by hand; also in capstans, windlasses and in clocks to prevent the recoil of a drum.

Fig. 63 shows a *retaining pawl* used to prevent rotation of the wheel in either direction; such pawls are used to retain pieces in definite adjusted positions.

Reversible pawls similar to the one shown in Fig. 64 are used in "feed mechanisms" upon planers and shapers. The pawl may be placed in the position A or the position A′; thus making it possible to feed the work in either direction. By the action of a triangular piece, with a spring and presser (all shown dotted) the

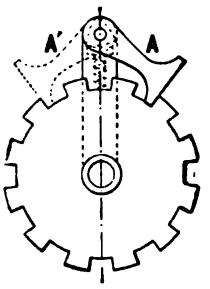


Fig. 64.

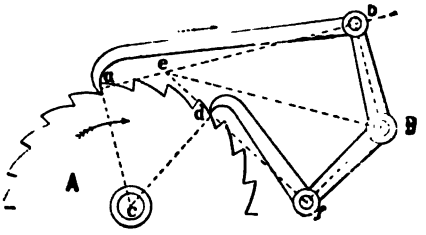
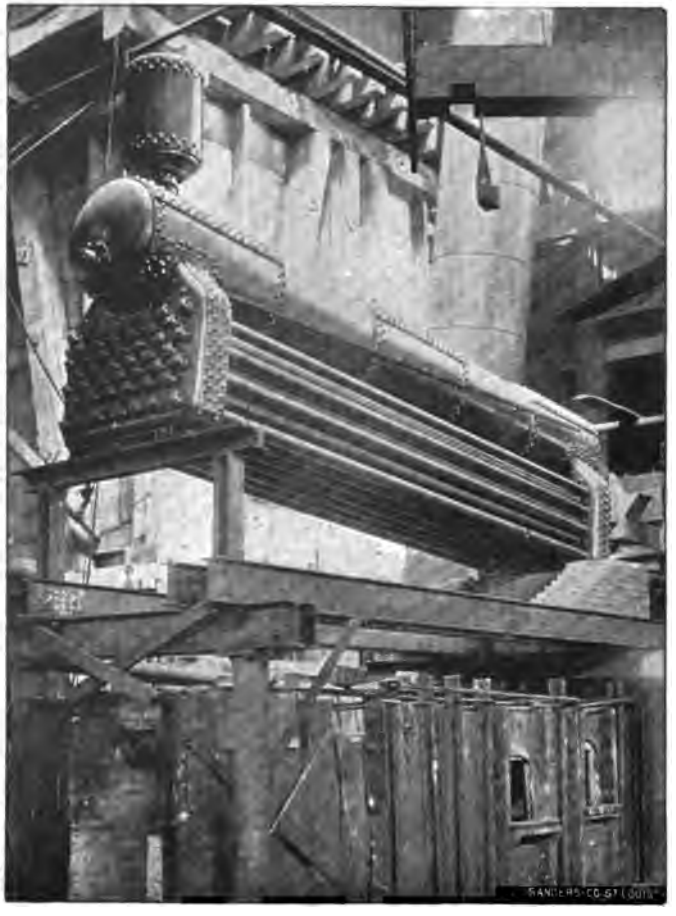
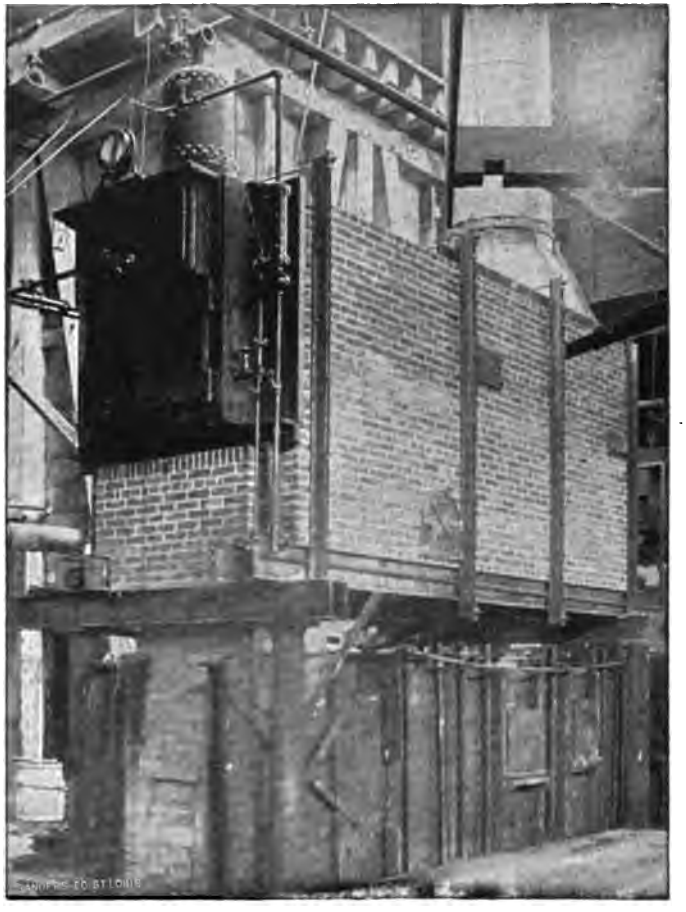


Fig. 65.

pawl may be held firmly against the wheel so as to feed in either direction, or if it is wished to throw the feed out of gear entirely, the pawl may be placed in a vertical position in line with the arm, and will be so held by the triangular piece.

Fig. 65 represents a ratchet wheel with two catches which pull the wheel instead of push it, as in the other cases. The relative positions of the catches, pivots and arms are clearly indicated. The lines of action *a b* and *d f* of the catches are perpendicular to the radii *c a* and *c d* respectively.





Views Showing Method of Setting a Heine Boiler for Utilizing Waste Heat from Puddling Furnaces, etc.

SECTION 3.

CONSTRUCTION OF BOILERS.

A steam boiler, or steam generator, consists of a vessel to contain the water and the steam after it is formed; a fire-box to contain the fire; tubes, flues and uptake to transmit heat and conduct the hot gases from the fire to the chimney, and various fittings to facilitate the safe and economical operation. Boilers are often classified according to their uses and conditions; thus we have stationary, marine and locomotive boilers. Boilers having a shell partially filled with tubes, through which the hot gases pass, are called tubular, fire-tube or shell boilers; and those having a large flue in which is placed the fire, are called flue boilers. If the tubes are filled with water and the hot gases are outside, the boiler is called a water-tube boiler.

Steam boilers are made in a variety of shapes, according to the type, uses and conditions. Let us first consider boiler construction in general, leaving out the peculiarities of marine, locomotive and water-tube boilers.

MATERIALS.

The materials of which boilers are constructed are exposed to conditions which weaken them and shorten the life of the boiler. Among these conditions are corrosion, both external and internal, high pressure, and expansion and contraction, due to varying temperature and pressure.

Cast iron was the material of which the earliest forms of boilers were made, but on account of its low tensile strength and its unreliable nature, it is now but little used, except for parts of water-tube boilers, and sometimes for the ends of low-pressure cylindrical boilers and for fittings. It is cheap and resists corrosion but on account of its unreliability and brittleness, the parts must be made thick and therefore heavy.

Wrought iron, up to about 1870, was the principal material used for boiler plates. It is a pure iron prepared from pig iron by a process called puddling, described in "Metallurgy." Wrought iron is well adapted for use in boiler construction, as it is strong, tough and fibrous, and combines high tensile strength with ductility and freedom from brittleness. When the properties mentioned are well combined, wrought iron will resist strains due to unequal expansion. Boiler fastenings, stays and other parts made by welding are sometimes made of wrought iron. It is customary to consider that a bar loses about one-quarter of its strength by welding, although it is often stronger in the weld, owing to the working of the metal during the welding process.

Steel has entirely displaced iron for boiler-shell work. Boiler steel is made by the open-hearth process, and contains for ordinary thickness of 1 or $1\frac{1}{4}$ inches 0.25 per cent carbon, while thinner plates of $\frac{1}{4}$ inch should not contain over 0.15 per cent carbon. Larger percentages of carbon, while accompanied by an increase in tensile strength, lessen the ductility. The following properties show steel to be the best boiler material at present: great tensile strength, ductility, homogeneity, toughness, freedom from blisters and internal unsoundness. Blisters and unsoundness are faults sometimes met with in wrought-iron plates.

Copper in many respects is superior to wrought iron for boiler construction. It is homogeneous, resists oxidation (the corrosive action of most feed waters) and incrustation. It is more ductile and malleable and a better conductor of heat, which not only gives it a higher evaporative power, but also enables it to last longer under the intense heat of the furnace. Its disadvantages are its low tensile strength, about 30,000 pounds per square inch, and its decrease of strength with an increase of temperature. In heating from the freezing point to the boiling point it loses 5 per cent of its strength, and at 550° F. it loses about one-quarter of its strength. For these reasons and on account of its high price, it is now seldom used in boiler work.

Brass is an alloy of copper and zinc in which the proportions of each vary considerably. The red color comes from a larger per cent of copper. Red brass is better and more expensive than yellow brass. Brass is used for valves, gauges and other fittings.

Bronze is an alloy of copper and tin, and is advantageously used for valves and seats of safety valves where the wear is great.

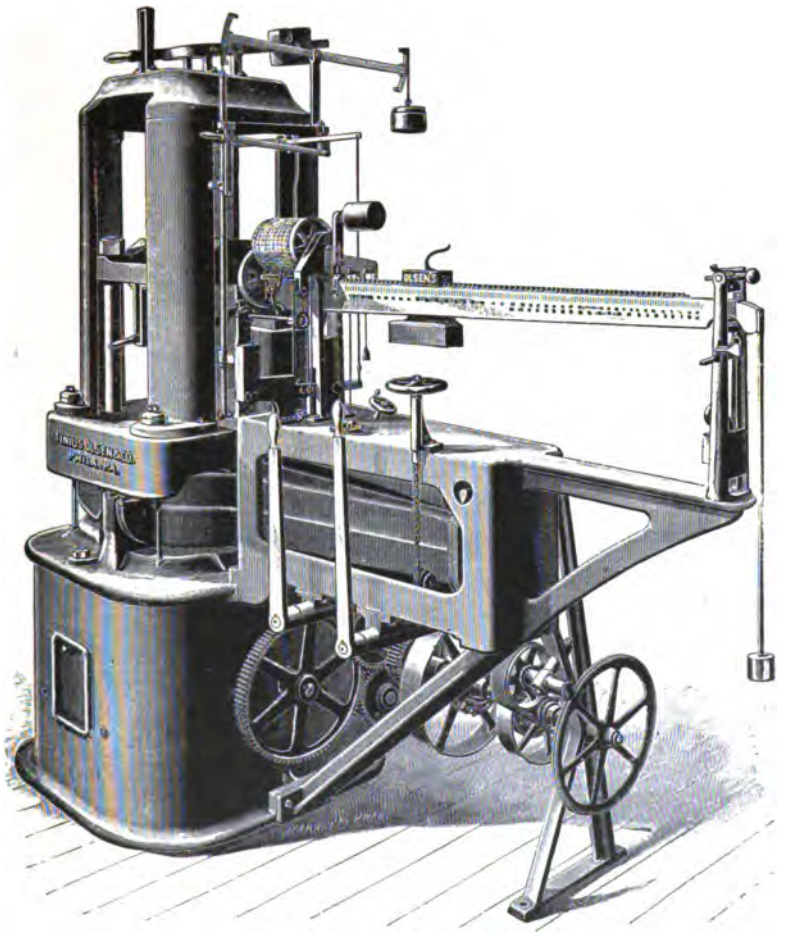


Fig. 1.

TESTING MATERIALS.

In order to determine the strength and the other qualities of the materials, specimens are tested. The results of these tests show the ultimate tensile strength, elastic limit, contraction of area a' d elongation.

The simplest way to test a piece of iron bar or plate would be to fix it firmly at the upper end and hang weights on the other end, adding other weights until the bar is broken. This is but a crude method, and in order that the elastic limit and elongation may be determined at the same time, testing machines are used. There is a large variety of testing machines, adapted for various materials, but the general principles are the same.

Testing Machines. The testing machine consists of a frame and two heads, to which the ends of the test piece are fastened by wedges or other devices. By means of steam or hydraulic power one head is drawn away from the other for tensile tests. The pull is transmitted to some weighing device, usually levers and knife edges like the beam of ordinary platform scales. In small machines the pull may be applied by a lever.

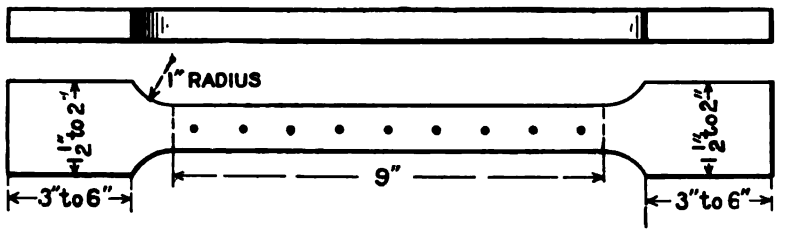


Fig. 2.

Testing machines are made for all varieties of testing: tensile, compressive and shearing stresses. Also for deflection of beams and for strength of wood, cement, brick and stone. Fig. 1 shows an Olsen testing machine designed for tensile and compressive tests of iron and steel.

In order to test materials, test pieces or specimens are prepared. For testing iron plate the test piece should be at least 1 inch wide, about 2 feet long and planed on both edges. Many engineers recommend these dimensions. According to the Board of Supervising Inspectors of Steam Vessels, the test piece should be 10 inches long, 2 inches wide and cut out at the center.

To ascertain the tensile strength and other qualities of steel, a test piece should be taken from each plate. These test pieces are made in the form as shown in Fig. 2. The straight part in

the center is 9 inches long and 1 inch wide; and to determine elongation it is marked with light prickpunch marks at distances 1 inch apart, the marked space being 8 inches in length. The ends are $1\frac{1}{2}$ inches to 2 inches broad and 3 inches to 6 inches long.

As has been explained in "Mechanics," the force necessary to break the piece is the proportionate part of the tensile strength per square inch. Thus if the test piece having a reduced section of .4 square inch is broken at 19,200 pounds, the tensile strength of the plate is $\frac{19,200}{.4} = 48,000$ pounds per square inch.

EXAMPLES FOR PRACTICE.

1. If a piece of boiler plate breaks at 33,500 pounds and the reduced section is $1\frac{1}{8}$ inches by $\frac{1}{2}$ inch, what is the ultimate tensile strength?

Ans. 59,555 pounds.

2. A boiler plate is claimed to be of 64,000 pounds tensile strength. If the section is 1 inch wide and .63 inch thick, what should be the reading of the testing machine when the specimen breaks?

Ans. 40,320 pounds.

3. A test piece of the form shown in Fig. 2 measured 8 inches between the prickpunch marks before testing and 9.56 inches after testing. What was the per cent of elongation?

Ans. $19\frac{1}{2}$ per cent.

4. If the area of section before breaking is .4825 square inch and after breaking is .236 square inch, what is the per cent of reduced area?

Ans. 51 per cent.

STRENGTH OF BOILER MATERIALS.

The crushing strength of cast iron is high, varying from 50,000 to 75,000 pounds per square inch; its tensile strength is low, varying with the chemical and physical properties of the iron from about 15,000 to 22,000 pounds per square inch.

Wrought-iron plates having a tensile strength of from 50,000 to 60,000 pounds, with an elongation or ductility of from 20 per cent to 30 per cent, are suitable for boiler work. Boiler iron may be

tested in the following ways if testing machines are not available: Cut from the plate a strip about 2 inches wide and bend it cold, down upon itself; if it shows no fracture on the outside curve, it is satisfactory. This is, however, a severe test, and only the best flange iron will stand it; on the other hand, any iron which, when heated to a cherry red and bent, shows cracks or fracture on the outer curve, is unfit for use in boiler construction. When wrought iron was used for boiler plates it was customary to give the plate what is called the hammer test. The plate was suspended clear of the ground and struck with a hammer at intervals of three or four inches over its surface; a clear, ringing tone indicating a sound plate, while a dull sound indicated with fair certainty a defect such as internal unsoundness.

Mild steel has a tensile strength of from 55,000 to 65,000 pounds per square inch, with an elongation of 25 per cent. A test piece cut from a plate $\frac{3}{4}$ inch thick or less should stand bending double, when hot or cold, and not show any cracks; thicker plates should be capable of being bent at a small radius to a large angle without showing any cracks. Steel should never be worked at a blue heat, as in this state it is very brittle. It is also mechanically tested by being heated to a cherry red, quenched in water at 82° F. then bent in a curve of small radius; if it cracks, it has become tempered, and it is therefore unsuitable for this work. If the tensile strength of the steel is under 70,000 pounds per square inch, it is sufficiently tough and ductile and can be easily worked.

In general, boiler materials are carefully tested for the following qualities:

Tensile strength, to resist rupturing strains. Also in order that the plates may be thin.

Toughness and elasticity, to resist corrosion and the wear and tear of manufacture.

Ductility, so that the boiler may change its shape slightly without rupture. This is a more important quality.

BOILER CONSTRUCTION IN DETAIL.

The drawing or design of the boiler is worked out in the draughting room, as explained later under the head of Boiler Design.

The draught shows the general arrangement of the boiler, together with complete detail drawings, from which the materials are ordered. These materials are plates, rods for stays, rivets, stay bolts, tubes, steel bars, angles and channel bars for stiffening, etc.

In some boiler shops it is customary to lay the boiler out on a large blackboard full size, thereby checking the drawing. In ordering plates the blank forms are filled out in the following manner:

Messrs. John Blank & Co:

Please furnish us with the following Steel Plates, Ultimate Tensile Strength, *60,000;* Elongation, *25 per cent:*

Number wanted.	Thickness	Dimensions.	Marks.	Remarks.
6	1″	90″×70″	S 14	Shell

The dimension which runs in the direction the plate is to be bent is given first. The plates are marked as per order blank, and this serves to identify the plate when the occasion arises. When ordering any odd shape, a sketch with dimensions must be placed in the column headed "Remarks."

In ordering plates, allow for trimming, particularly in the case of irregular shapes. Rivets are sold by the pound, regardless of their shape or size. Round and flat iron may be ordered by the running foot. Manufacturers publish tables showing weight of rivets, round iron, etc., with which they furnish boiler makers.

Boiler shops are equipped with the following tools: plate rolls, plate planers, shears, drill presses, punches, countersinking machines, flanging machines, hydraulic and steam riveters, and a compressed-air system for operating pneumatic machines, such as calkers and chippers. They also have machine shops for doing

such machine work as is required for fittings, furnace fronts, etc., and a system of cranes for handling and transporting material. In connection with the above is a storeroom of sufficient size, a forge shop, and an engine and boiler for supplying the shop with the power necessary to operate it.

In boiler-shell work drilling has entirely displaced punching, and to-day all holes are drilled. Punching is cheaper than drilling, but it is more injurious to the plates and not as accurate. It is easy to see that drilling rivet holes, even if twenty are being drilled at once, is done with less strain on the plates than when done by a multiple punch forcing several holes at once. The force required to punch a plate gives the best idea of the harm done to the plate. Experiment shows that the resistance of a plate to punching is about the same as its resistance to tensile tearing. Suppose this to be 50,000 pounds per square inch; then the force required to punch the plate is the area cut out times the shearing strength, or $d \times \pi \times t \times 50{,}000$.

In which formula

$d =$ diameter in inches and

$t =$ thickness in inches.

For a hole $\frac{3}{4}$ inch in diameter in a $\frac{1}{2}$-inch plate, the force will be

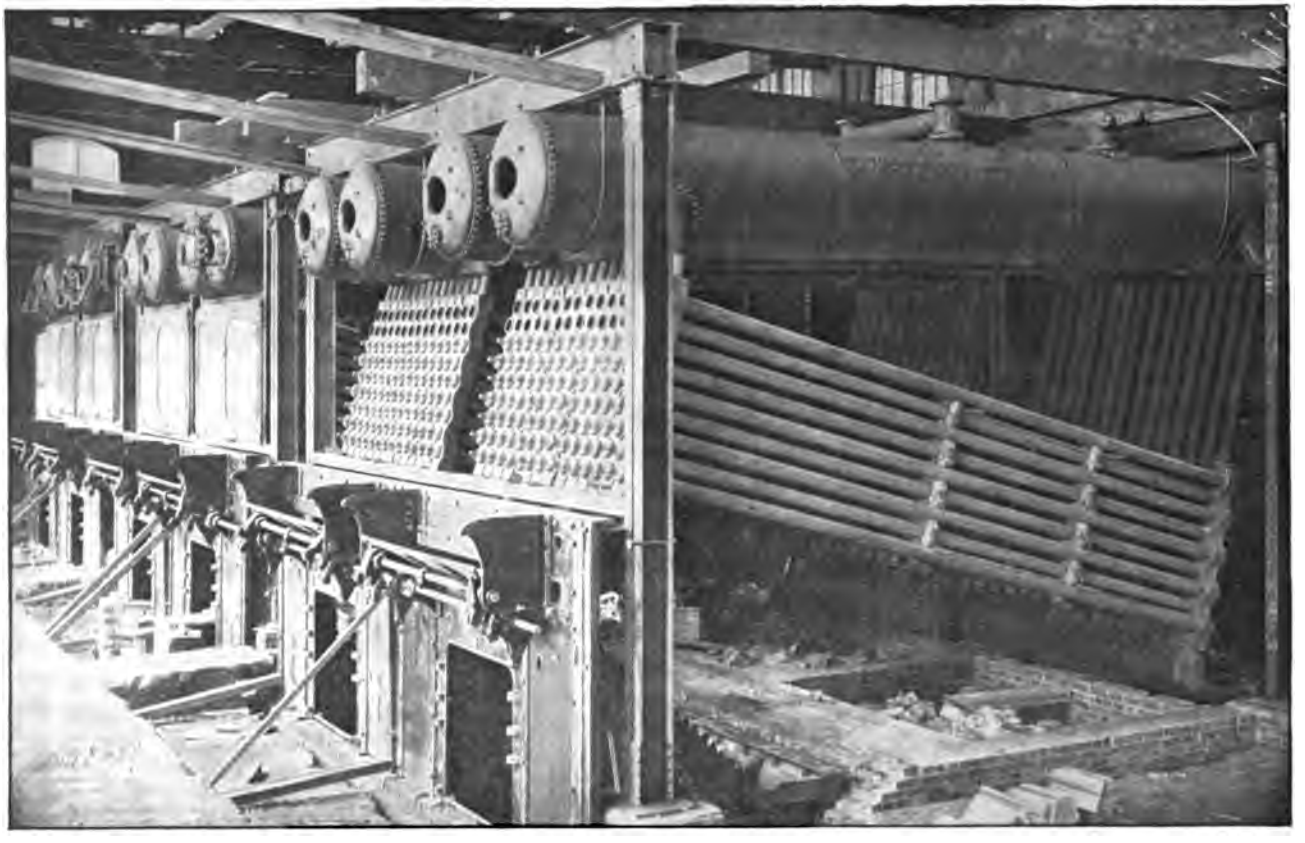
$$\tfrac{3}{4} \times 3.1416 \times \tfrac{1}{2} \times 50{,}000 = 58{,}900 \text{ pounds.}$$

If the force required to punch one hole is 58,900 pounds, the force required in punching several holes by means of a multiple punch is enormous.

A good, ductile plate is but little injured by punching; but if of a hard, steely nature, it is likely to be seriously injured. For this reason wrought-iron plates are usually punched and steel plates are drilled. On the whole, a drilled plate is somewhat stronger than a punched plate for any kind of joint.

Some boiler makers punch the rivet holes slightly smaller than the desired size and then ream them out. By this process the injured metal around the holes is cut away. Another method to overcome the injurious effects is to anneal the plate after punching.

The ordinary process of annealing consists of heating the plate to red heat, and then allowing it to cool slowly. By this



1600 Horse Power Babcock and Wilcox Boilers in Process of Erection.

means, hard and brittle iron or steel is made soft and tough. While the metal is hot, the surface becomes oxidized. For most purposes this scale of oxide in not harmful, but in some cases it must be removed. As this is expensive, a process of annealing in illuminating gas has been devised. The action of the gas is to reduce the oxide without altering the properties of the piece. The results obtained from annealing depend upon the kind of iron or steel, the temperature to which it is raised, and the rate of cooling. It is a great advantage to all steel of over 64,000 pounds per square inch in tensile strength, but softer steels are little better for the process.

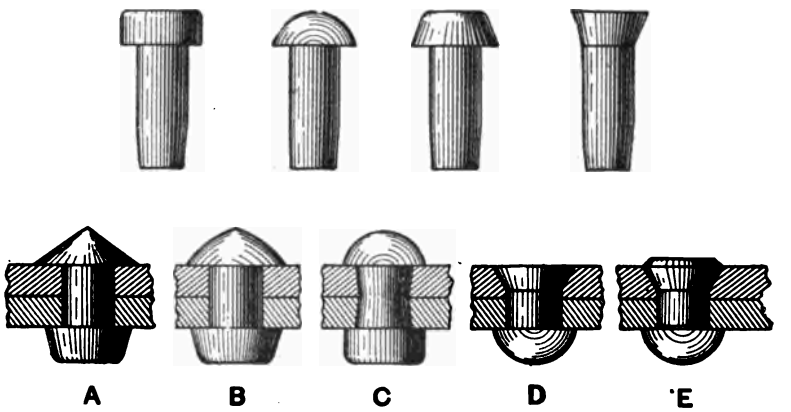


Fig. 3.

After the shell plates are planed to correct shape and the holes drilled or punched, they are put through the bending rolls and bent into a cylindrical shape, the amount of curvature being determined by a template made for the purpose. Plates are usually sheared to size, and then the edges planed with a slight bevel to facilitate calking. In the meantime the heads are being flanged by a hydraulic flanging machine; when the flange is completed, the head is put on the platen of a boring mill and turned so as to exactly fit into the shell. In some shops it is customary to punch or drill only a few holes in the shell and flange of the head, these holes serving to take bolts for holding the parts together. The back head plate is bolted into the rear course of plating, and

the parts thus assembled are hoisted up to drill if the plates, etc., have not been previously drilled or punched, otherwise to the hydraulic riveter.

RIVETS AND RIVETING.

Rivets are formed by forging, from round iron bar or mild steel, with a cup or pan shaped head. The cylindrical part, called the shank, is a little smaller than the hole and has a slight taper. Fig. 3 shows common forms of rivets. As rivets are not as reliable in tension as in shear, they are used mainly at right angles to the straining force. If the stress is parallel to the axis, bolts are used, since they are strong in tension. The shearing strength of steel rivets is about 45,000 pounds per square inch, and of iron rivets about 40,000 pounds per square inch. Steel rivets are often used with steel plates, but many boiler makers prefer to use iron rivets in all cases.

Three types of rivets in use are shown in Fig. 4, the following table giving the dimensions:

Diameter of Rivet.	Cone Head. A			Countersunk. B		Button Head. C	
D	E	F	G	E	G	E	G
$\frac{5}{8}$	$1\frac{1}{16}$	$\frac{1}{32}$	$\frac{9}{16}$	$1\frac{1}{16}$	$\frac{9}{32}$	$1\frac{1}{16}$	$\frac{7}{16}$
$\frac{11}{16}$	$1\frac{1}{8}$	$\frac{21}{32}$	$\frac{39}{64}$	$1\frac{3}{16}$	$\frac{5}{16}$	$1\frac{1}{8}$	$\frac{1}{2}$
$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{23}{32}$	$\frac{21}{32}$	$1\frac{1}{4}$	$\frac{3}{8}$	$1\frac{1}{4}$	$\frac{9}{16}$
$\frac{7}{8}$	$1\frac{7}{16}$	$\frac{13}{16}$	$\frac{3}{4}$	$1\frac{3}{8}$	$\frac{7}{16}$	$1\frac{7}{16}$	$\frac{3}{4}$
1	$1\frac{5}{8}$	$\frac{15}{16}$	$\frac{27}{32}$	$1\frac{5}{8}$	$\frac{1}{2}$	$1\frac{5}{8}$	$\frac{3}{4}$

Formerly all joints of boilers were riveted by hand, but now all riveting is done by machines, except those joints to which a machine cannot be applied. If done by hand, the red-hot rivet is inserted in the hole, and the second head formed by two riveters working with hammers. This head is either made conical by the hammers alone or finished with a cup-shaped die called a "snap." This latter is the more usual method. The disadvantages of hand riveting are slowness and a tendency to form a shoulder before the rivet fills the hole.

Machine riveting is preferable, as the work is done better, faster and more accurately; the pressure coming gradually on the entire rivet, compresses it completely into the hole before the head is formed. Before riveting, care should be taken that the plates are close together, so that a shoulder will not be formed between the plates and prevent a good joint. Rivets should always be put in while red hot, for in this condition they are more easily worked, and when they cool they contract, nipping the plates together in a tight joint.

Hydraulic riveting is more gradual and is generally preferred to steam riveting. The pressure from the steam riveter often comes as a sudden blow and does not allow time for the rivet to completely fill the hole.

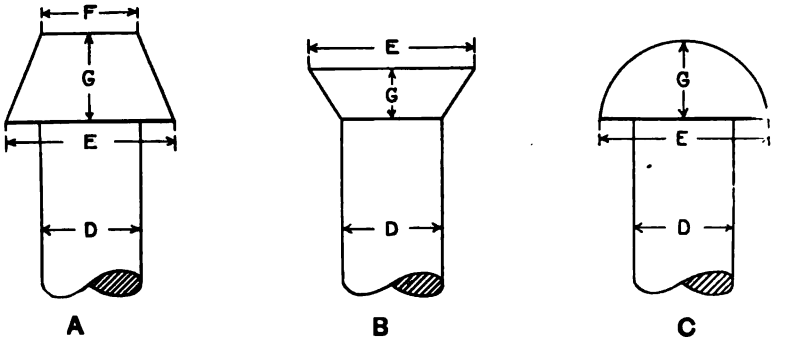


Fig. 4.

It is sometimes desirable to rivet with a countersunk head; that is, the rivet does not project above the plate. The countersunk head is formed by hammering down the end of the rivet into the countersink in the plate. This form is shown at D, Fig. 3. This joint is often used in shipbuilding and in boiler making when it is necessary to attach mountings. It should always be avoided, if possible, on account of its weakness, and especially when the straining force acts in the direction of the length of the rivet, as the head has a very insecure hold and is likely to be pulled through the hole.

Rivets may be tested in a boiler shop as follows: the rivet to be bent cold in the form of a hook around another rivet of the

same diameter, and show no flaws or cracks; to be bent hot down upon itself and show no cracks, head to be flattened while hot until its diameter is $2\frac{1}{2}$ times the diameter of the shank, and show no flaws.

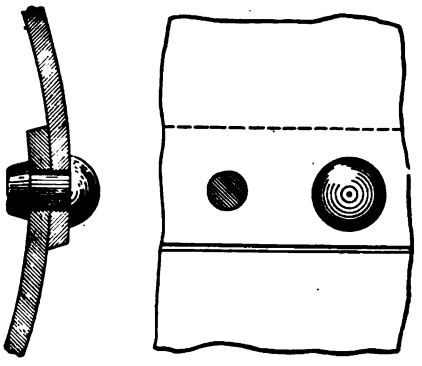


Fig. 5.

The uniform heating of steel rivets is of more importance than in the case of iron rivets, where it is sufficient to heat the points only. Steel rivets also should not be heated to a white heat, as iron rivets are, but to a bright cherry red, for if heated beyond this point they will burn. The fire in which steel rivets are heated should be kept thick, and the draught moderate. This should also be observed in heating steel plates for flanging.

There are various forms and strengths of riveted joints. It

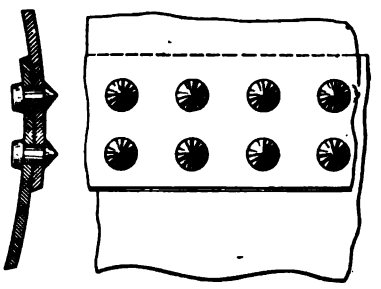


Fig. 6.

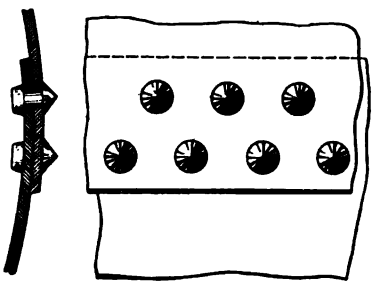
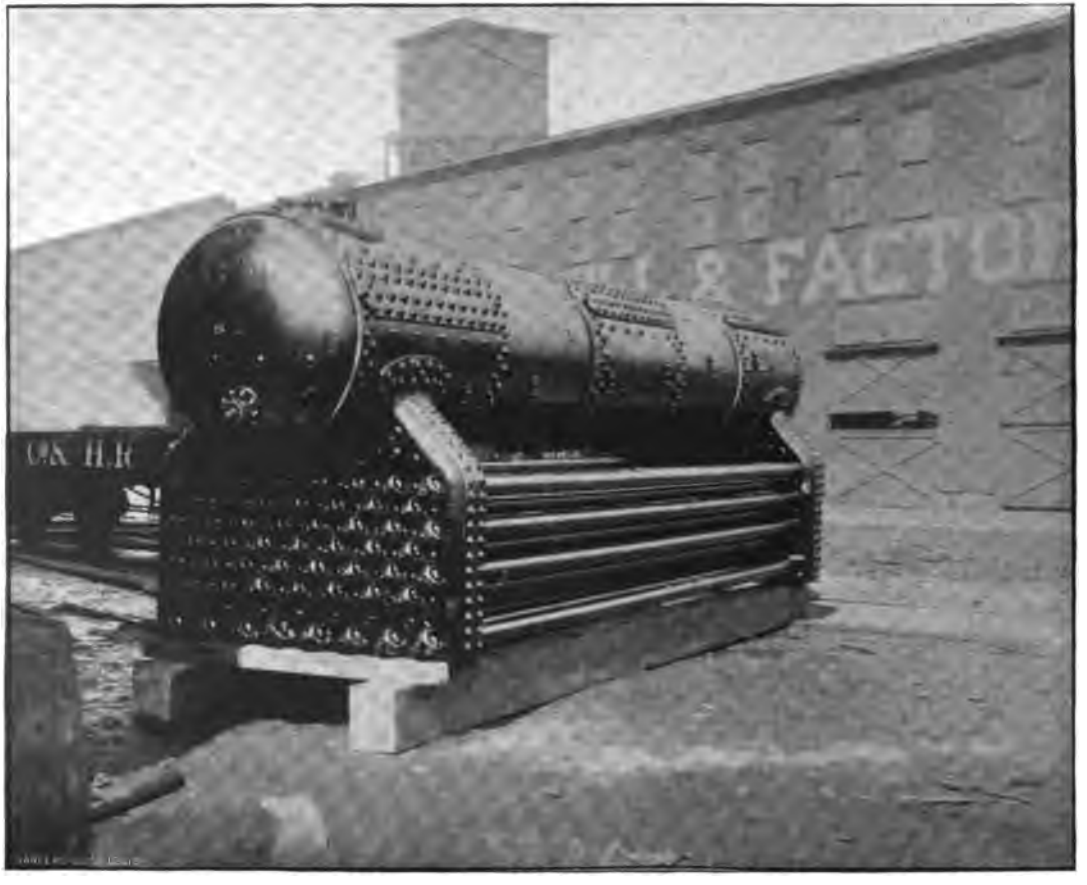


Fig. 7.

is obvious that in punching or drilling, a plate is weakened to the extent of the sectional area cut out, and that if the holes are punched, the metal between the holes is weakened. In treating the strength of a joint it is customary to speak of it as a percentage of the strength of an unpunched plate.

If one plate overlaps another and is riveted to it by a single



100 Horse-power Heine Boiler, Designed for 400 Pounds per Square Inch Working Pressure.

row of rivets, as shown in Fig. 5, it is called a single-riveted lap joint. This joint has about 56 per cent of the strength of a solid plate. If another row of rivets is added, it is called a double-riveted lap joint; Fig. 6 shows the double-riveted lap joint chain riveted, and Fig. 7 the double-riveted lap joint zigzag riveted.

Double riveting is done in two ways: zigzag, or staggered, and chain. When rivets are put in so that the rivets of one row are opposite the spaces of another row, it is called zigzag riveting or staggered riveting. If the rivets are placed immediately opposite each other, it is called chain riveting.

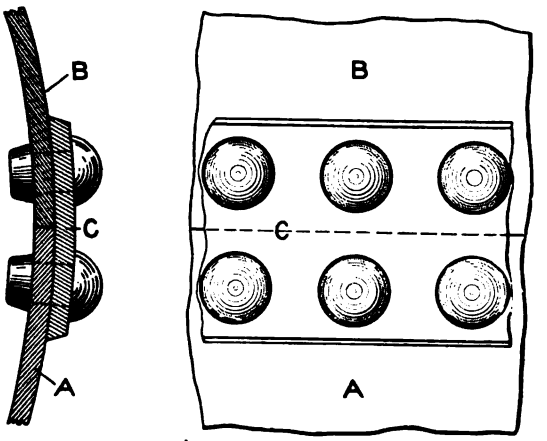


Fig. 8.

If the two plates are kept in the same plane and a cover or butt strap riveted on, it is called butt riveting (Fig. 8, in which A and B are the boiler plates, and C is the butt strap). If an inside butt strap is added, it is called a double butt joint (Fig. 9). Fig. 10 shows a treble-riveted butt joint. A single butt joint is about equal in strength to a lap joint having but one row of rivets, but a double butt joint is considerably stronger.

In this latter form of joint the rivets have double shearing surfaces, since they tend to shear off in two planes. This either makes a stronger joint or allows the use of smaller rivets. In the single butt joint the butt strap is usually about $1\frac{1}{8}$ the thickness of the plate, and if the inside butt strap is added, each butt strap

is made about $\frac{5}{8}$ the plate thickness. Butt joints are now being used in the best class of boilers, and are used almost entirely for plates less than $\frac{1}{2}$ inch in thickness.

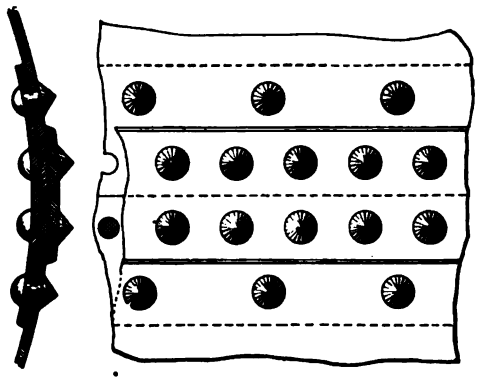


Fig. 9.

Lap joints are used for circumferential seams, and the stronger joint, the butt, for longitudinal joints. For high pressures in marine boilers, triple riveting is frequently used.

If a cover plate is riveted on the outside of a lap joint, it is called combined lap and butt joint. In this case there are three rows of rivets, the middle row having twice as many rivets as the outer rows. Fig. 11 shows the combined joint.

The distance between the centers of rivets is called the "pitch." The mathematical calculation of pitch and the distance between the rivets and the edge of the plate will be taken up later.

The following table gives an idea of the relative strengths of riveted joints:

Kind of Joint.	Riveting.	Percentage of Strength.	
		Punch.	Drilled.
Lap	Single	55	62
	Double	69	75
Single Butt	Single	55	62
	Double	69	75
Double Butt	Single	57	67
	Double	72	79

FLANGING IRON AND STEEL PLATES.

Iron plates are more severely tested by flanging than by any other work done upon them. This is due to their fibrous nature, and great care is necessary to prevent breaking in the bend, if the corner is sharp.

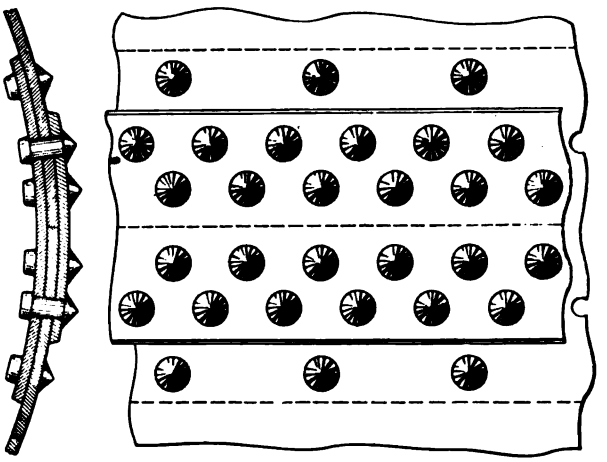


Fig. 10.

As has been stated, steel requires uniform heating and moderate curves. Flanging is almost entirely done to-day by machines. After flanging, the steel should be annealed by heating the whole plate uniformly to a dull red heat, and allowing it to cool slowly.

WELDED JOINTS.

Welded joints for boiler shells are desirable. By their use deposits which accumulate on and around rivet heads and joints, corrosion caused by leakage, and loose rivets, are done away with, and calking also. Moreover, a perfectly welded joint is stronger than the best riveted joint, and approximates nearly to the original strength of the plate. Welded steam drums are used now quite extensively for water-tube boilers of the marine type.

The soundness of such a joint is a matter of uncertainty, and

depends upon the skill and care of the workmen. It is impossible, from external appearances, to judge the soundness of a welded joint. The principal use of welded joints is for furnace tubes and steam domes, but they have not been used much for boiler shells. The lack of tests on welded joints and the small amount of information on the subject, render the results of experiments of little value. The weld is best made when the edges of the plates are upset, at red heat, to nearly double the plate thickness, and beveled to an angle of about 45 degrees. The edges are then heated together, and the weld made by hammering down the joint to the original thickness of the plate.

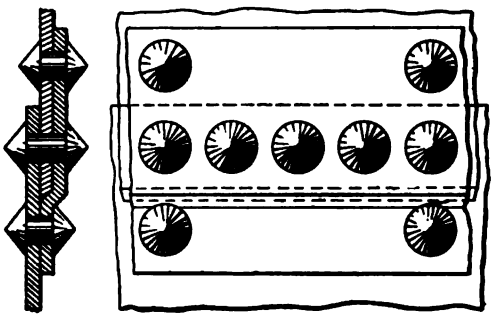


Fig. 11.

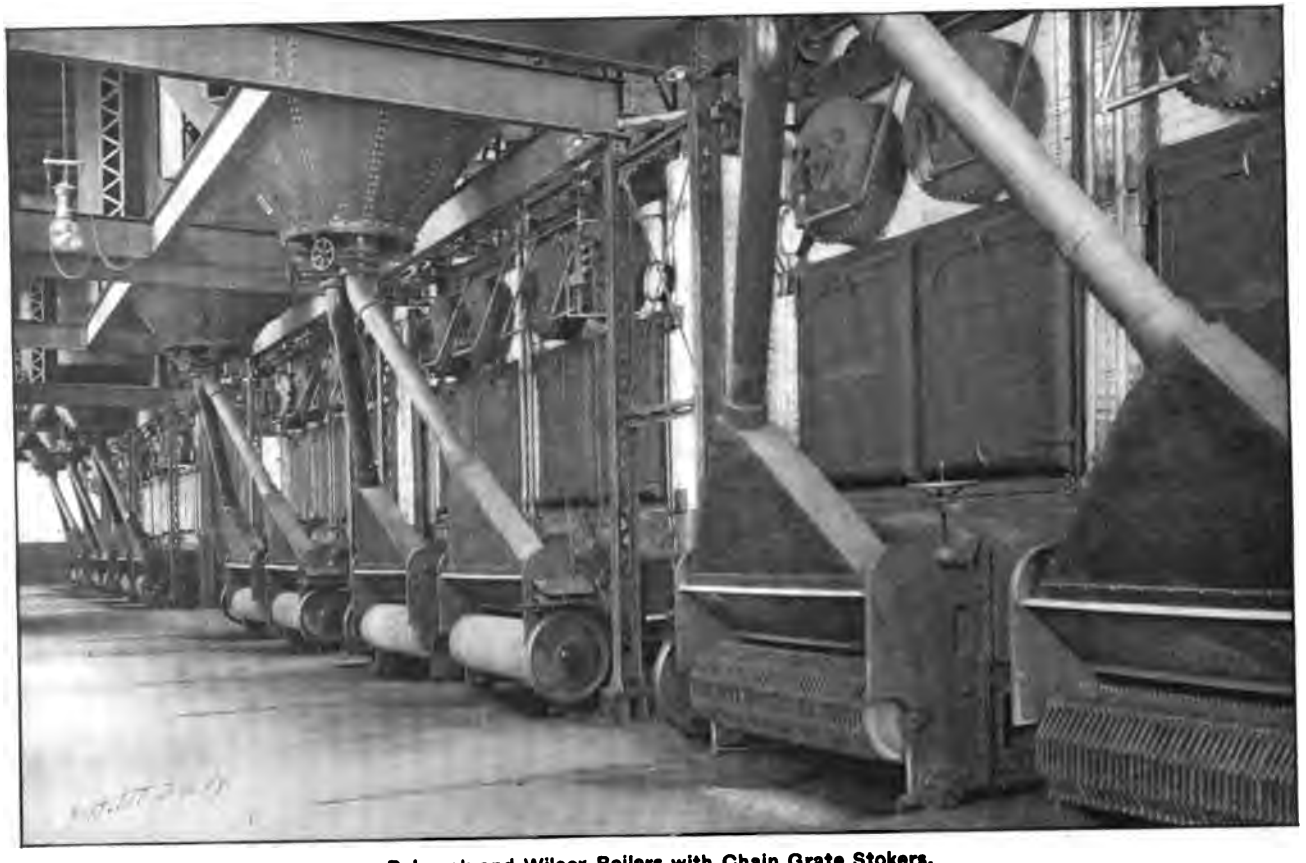
ARRANGEMENTS OF PLATES AND JOINTS.

When we take up the design of boilers we shall see that a boiler tends to rupture longitudinally. The reason for this is that the resistance of a thin cylinder to circumferential rupture is double the resistance to longitudinal. Since this is the case, lap joints are used for transverse seams, and a stronger form (the double butt joint) is used for the longitudinal.

At the junction of three or more plates, where the circumferential and longitudinal joints meet, ordinary riveted joints would be too thick. To overcome this difficulty, two or more plates are forged thin at the joint, as shown in Fig. 12.

Whenever longitudinal and girth seams meet, the plates should be arranged to "break joints"; that is, one longitudinal seam should not be a continuation of another. The proper arrangement is shown in Fig. 13.

In both vertical and horizontal boilers the inside lap is made to face downward, so that it will not form a ledge for the collection of sediment.



Babcock and Wilcox Boilers with Chain Grate Stokers.

The belts of plates that make up the length are sometimes arranged conically, with the outside lap facing backward. When the boiler is slightly inclined toward the front end, this conical arrangement facilitates draining and cleaning, as the dirt is removed at the front end. This is a great advantage to internally fired boilers, as they are difficult to clean.

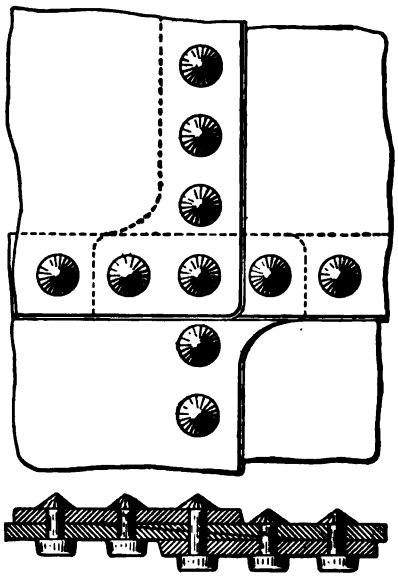


Fig. 12.

In long vertical boilers the ring seams are arranged with the inside lap facing downward, so as not to have a ledge for sediment. Sometimes the belts of locomotive boilers are arranged telescopically, with the largest diameter at the fire-box end. Of late years the best makers use larger plates than formerly. This is advantageous, especially in externally fired multitubular boilers, as the single seam is placed above the water-level, and therefore is away from the fire.

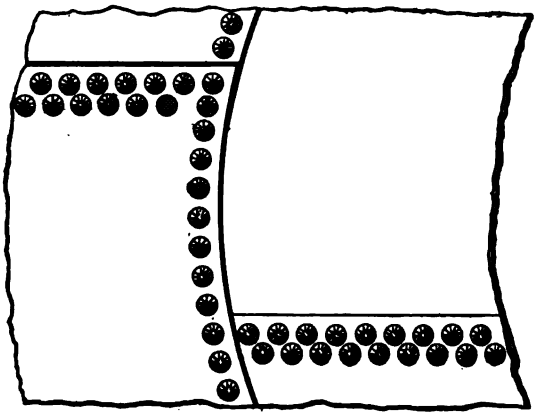


Fig. 13.

The portion of a boiler between the shell and the furnace is called the water leg. Figs. 14 to 20 inclusive illustrate the method of construction of the water leg and the joints around the furnace door. Figs. 14 and 15 show two methods of constructing

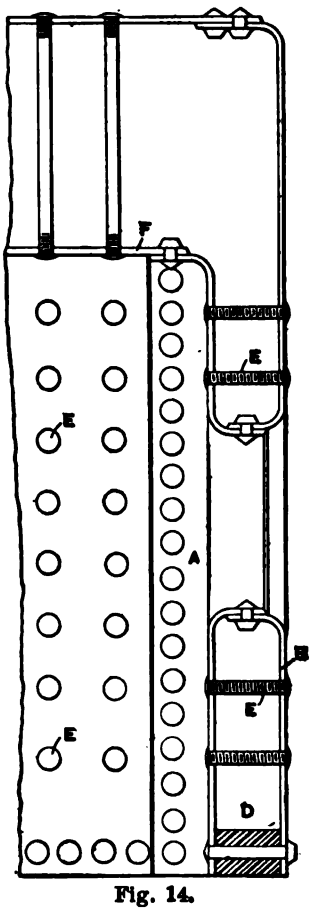


Fig. 14.

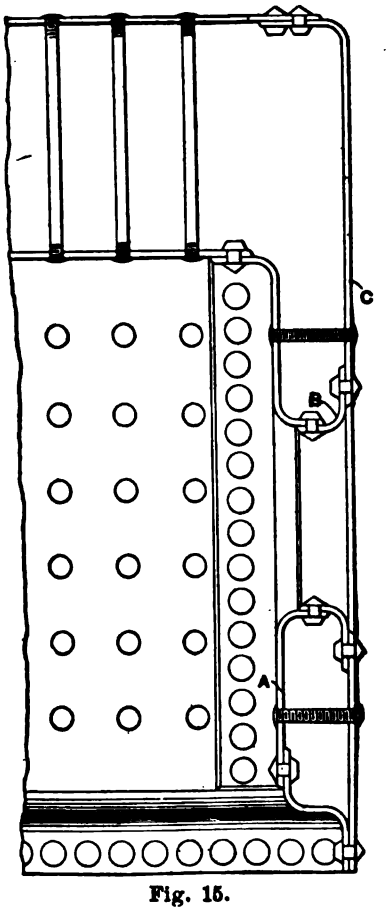


Fig. 15.

the water leg. In Fig. 14 the exterior plate and the furnace plate are riveted to the ring D by means of long rivets. This ring is usually made of wrought iron, but in many cheap boilers it is of cast iron. In Fig. 15 the two plates are riveted to the flanged ring D. This construction is better than the solid cast-iron ring, on account of flexibility, but the junction of the plates D and C

forms a corner in which sediment is deposited. In Fig. 17 the plate B is flanged and riveted to C. This arrangement requires less riveting than the one shown in Fig. 15. Figs. 14, 15 and 17 also show three forms of construction of the joints around the

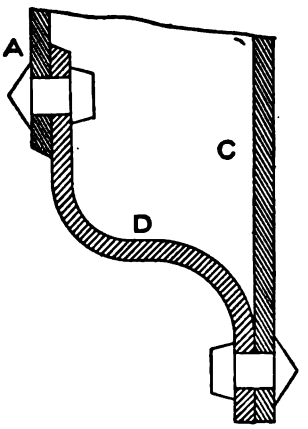


Fig. 16.

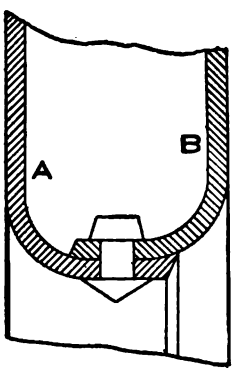


Fig. 18.

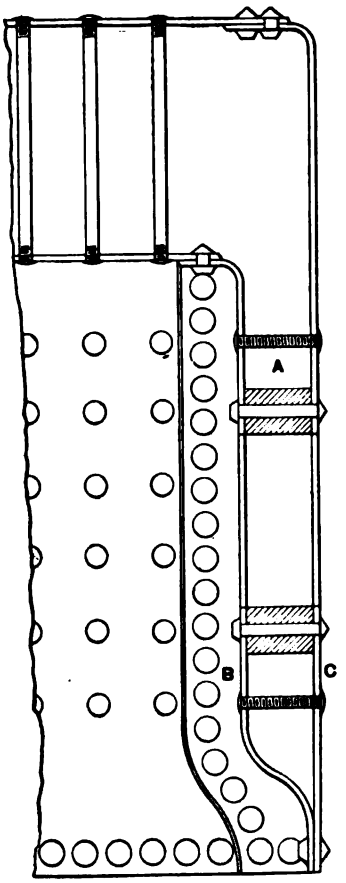


Fig. 17.

furnace door. In Fig. 14 both the exterior plate and the furnace sheet are flanged and riveted together. This is shown in an enlarged view in Fig. 18. The construction shown in Figs. 15 and 19 is not as good as that in Fig. 14, because of the extra riveting; also, it has two corners, B and C, for the deposit of sediment. Fig. 17 shows a somewhat different form of furnace construction,

the two plates being riveted to the cast-iron ring. This form is better shown in Fig. 20. It makes this part of the boiler too rigid, but it has the advantage of not having rivet heads to wear off. In these methods of riveting, those which have the flanged ring are preferable to those using the cast-iron ring, because of more freedom for expansion; but the flanged ring forms an undesirable corner.

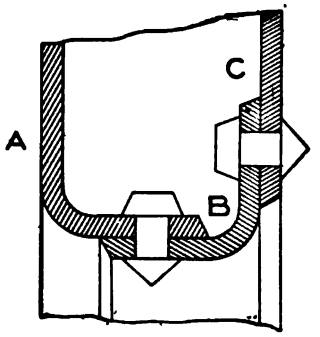


Fig. 19.

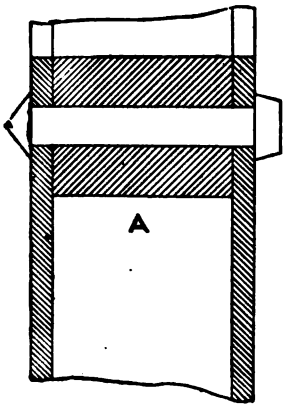


Fig. 20.

In almost every boiler, plates must be connected at right angles. An example of this is seen where the end plates are jointed to the shell plates of cylindrical boilers. There are three principal methods: riveting both plates to an angle iron, riveting to a flanged ring and flanging the end plate. In Fig. 21 the two plates are riveted to an angle iron, which is made of wrought or cast iron. This construction is too rigid; the constant variations of temperature cause repeated changes of form, which tend to crack the angle iron on the inside of the plate at the joint. Corrosion increases the evil, as it rapidly attacks iron which has once been cracked or broken. There is no definite rule for the dimensions of these angle irons, but it is safe to make the mean thickness a little greater than that of the plates.

The forms shown in Figs. 22 and 23 are better. The head is flanged and riveted to the shell plates. The flanging makes a more flexible joint. The radius of the curve of the flange should be about four times the thickness of the plate. The head and

shell are sometimes connected to a flanged ring, as shown in Fig. 24. The extra row of rivets makes a complex joint.

In vertical boilers the external fire-box is joined to the cylindrical shell by riveted joints. Figs. 25 and 26 show two forms; that in Fig. 25 being the better on account of the flanged ring,

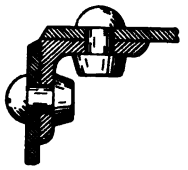


Fig. 21.

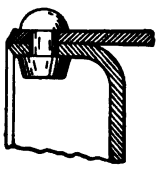


Fig. 22.

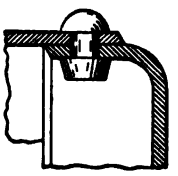


Fig. 23.

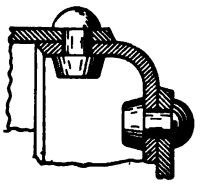


Fig. 24.

which allows expansion and contraction of the shell and furnace plates.

Sometimes the case occurs of connecting two plates which are parallel and near together. For instance, at the bottom of the

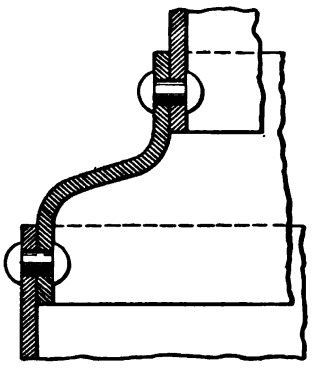


Fig. 25.

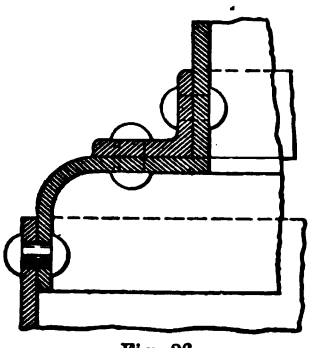


Fig. 26.

locomotive fire-box a connection must be made between the inner and outer fire-box. The water-leg construction is a similar case. Several methods for this construction are shown in Fig. 27. Fig. 27A is too complicated and is undesirable, both on account of the numerous rivets and angle irons, and on account of the inside joints, which cannot be calked. Fig. 27B is better, since it has but one angle iron; it has, however, the undesirable inside joint.

Fig. 27D is a good joint, the form of connection being called a channel iron. Fig. 27E, as we have seen, is a good flexible joint, but it has the undesirable corner where sediment lodges.

We have thus briefly discussed the various methods and arrangements for putting shells together, and now let us return to our boiler, which is ready for riveting at the hydraulic riveter. A few rivets are first driven at equal intervals around the ring seam

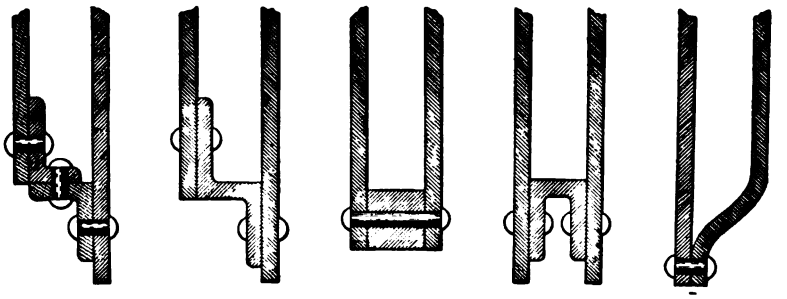


Fig. 27.

at the back head. The reason for driving only a few rivets is that any errors in the spacing of the holes are distributed and not accumulated, as would be the case if they were driven in succession. From this point on, the riveting is continued until the shell is completely riveted up.

STAYING.

The shell is now ready to receive the stays. When under steam, a cylindrical shell is strained by internal pressure in two directions, namely: transversely, by a circumferential strain due to the pressure tending to burst the shell by enlarging its circumference, and longitudinally, by the pressure on the ends. If a boiler were spherical it would require no stays, because a sphere subjected to internal pressure tends to enlarge but not to change its shape. All flat surfaces in boilers must be stayed, otherwise the internal pressure would bulge them out and tend to make them spherical in shape. The ends of steam drums on high-pressure water-tube boilers are often made hemispherical.

The first and most important point in staying is to have a

sufficient number of stays so that they will entirely support the plate without regard to its own stiffness. The second is to have them so placed as to present the least obstruction to a free inspection, and third, to have them so arranged as to allow a free circulation of water. Too much care cannot be taken in fitting stays and braces, as they are out of sight for long periods, and a knowledge of their exact condition is not always easily obtained. In the ordinary fire-tube boiler the principal surfaces stayed are: the flat ends, crown sheets, flat sides of locomotive boilers and combustion chambers of cylindrical marine boilers. In the case of most marine or Scotch boilers, the diameter is large compared to the length; hence the flat surface is considerable, and needs careful staying. All the plates that are not cylindrical or hemispherical must be stayed. The details should be arranged for each boiler; a few general methods and cautions can, however, be given.

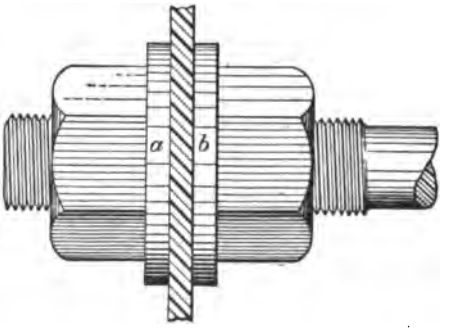


Fig. 28.

The most common and simple form of stay is a plain rod. It is used to stay the flat ends of short boilers. This stay is a plain

Fig. 29.

rod passing through the steam space and having the ends fastened to the heads. The ends are fastened and the length adjusted in a variety of methods; the simplest being nuts on both sides of the plate, as shown in Fig. 28. The copper washers *a* and *b* strengthen the plate and prevent abrasion by the nuts. In place of the nuts the rod is often bolted to angle irons, which are riveted to the plates. In this case, turn buckles similar to the one shown in Fig. 29 are used for adjusting the length.

The stays are usually from $\frac{3}{4}$ inch to an inch in diameter, and are made of wrought iron or steel, with an allowable stress of 5,000 to 7,000 pounds per square inch. If the ends are fastened to riveted angle irons, the combined area of the rivets is made a little greater than that of the rod.

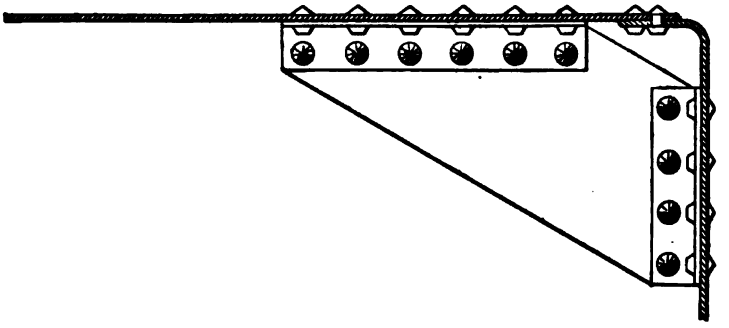


Fig. 30.

If a boiler is long, that is, more than 20 feet, long stays would sag in the middle and not take up the full stress on the end plates. For long boilers, gusset and diagonal stays are used. This form of boiler stay, shown in Fig. 30, is made of wrought-

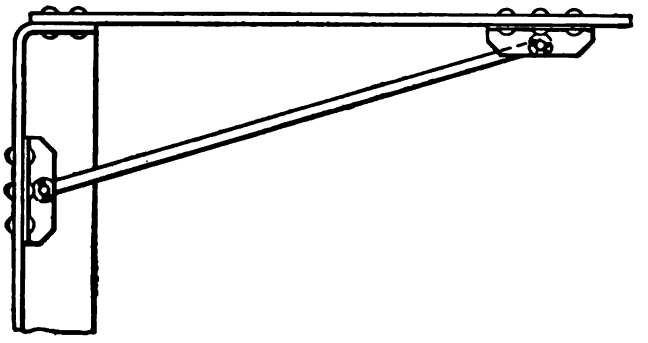


Fig. 31.

iron plate riveted to angle irons; the angle irons being riveted to the end and shell. Boilers of the Cornish, Lancashire and Galloway types often have this kind of stay. These boilers are internally fired, and as the variation of temperature causes expansion and contraction, great care should be used in placing the gusset

stay. If the stay is too near the flange or too many stays are used, the head will be too rigid and have a tendency to crack.

A form of diagonal stay is shown in Fig. 31. The plain rod is connected to angle irons by means of split pins. The angle irons are fastened to the shell and end by rivets or bolts. Another form of diagonal stay, called the crowfoot, is shown in Fig. 32. The two ends are bolted or riveted to the end and shell.

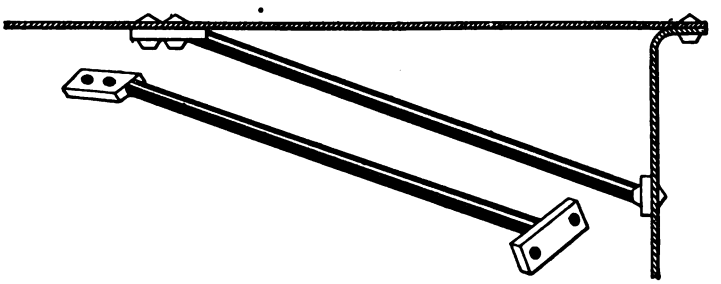


Fig. 32.

The angle between the shell plate and stay rod should be small,— not more than 30 degrees. The rod itself is designed for tensile strength, since the diagonal pull may be easily reduced to an equivalent direct pull. A large factor of safety is used to provide for future corrosion.

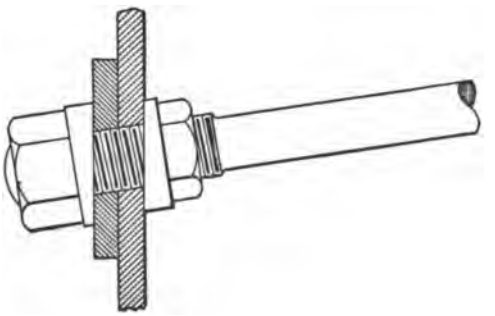


Fig. 33.

For marine boilers, a modified crowfoot stay (Fig. 33) is often used. The end passing through the head is supplied with nuts and taper washers, the washers having the proper taper to allow the nuts to be set up tightly against them.

In locomotive fire-boxes and in the combustion chamber of marine boilers, there are two flat or slightly curved surfaces that must be stayed together. These are riveted by short screw stay bolts. The bolts shown in Figs. 34 and 35 are screwed in place, and the ends riveted over. In marine boilers these stays are fastened with nuts, as shown in Fig. 36, instead of being riveted.

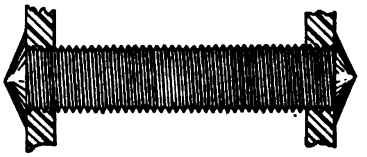


Fig. 34.

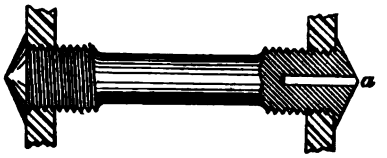


Fig. 35.

Sometimes the bolt is threaded the entire length, as in Fig. 34, or is turned off smooth in the center, as in Fig. 35. The smooth surface resists corrosion, and is less likely to fracture than the threaded bolt. Sometimes a small hole is drilled in the end, so that if the bolt breaks, the escaping steam will give warning. This is shown at *a*, Fig. 34. These bolts are $\frac{7}{8}$ inch or 1 inch in diameter.

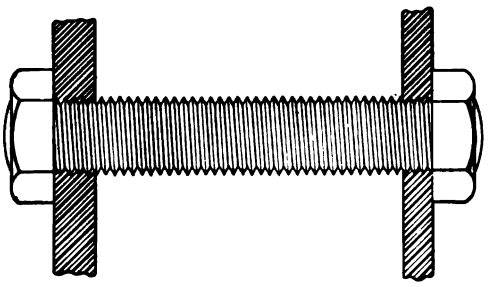


Fig. 36.

The strains which come on a stay bolt are not the same as those on rivets or on ordinary stay rods; as a matter of fact, stay bolts fail by a bending stress, and generally fracture just inside the outside sheet, due to the unequal expansion between combustion chamber or furnace and the outside boiler shell. Owing to this difference of expansion, flexible stay bolts have been designed, but have not come into general use, nor are they likely to, as they occupy considerable space and are much more complicated than the simple stay bolt. Stay bolts are made from the best quality of refined iron, which has been found to stand the strains of alternate heating and cooling better than mild steel. Iron stay bolts are more durable, because of the fibrous nature.

It should be added that boiler heads are further stiffened by channel bars or angles placed along the line of holes for the through stay rods.

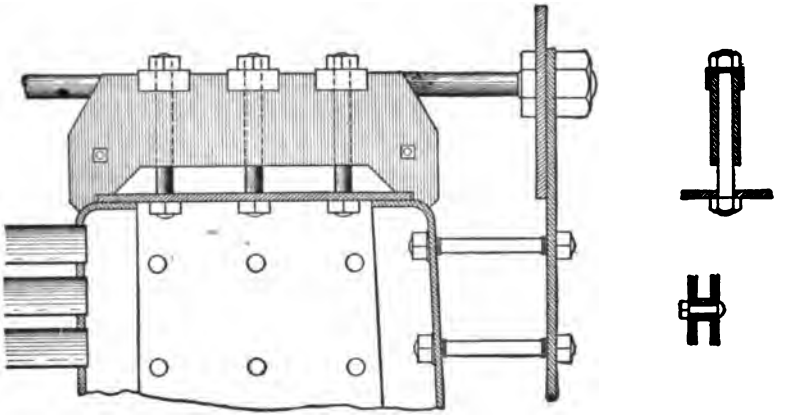


Fig. 37.

The crown sheets of fire-boxes and tops of combustion chambers are usually stayed by crown bars, which extend across the flat surfaces, as shown in Fig. 37, the ends resting on the

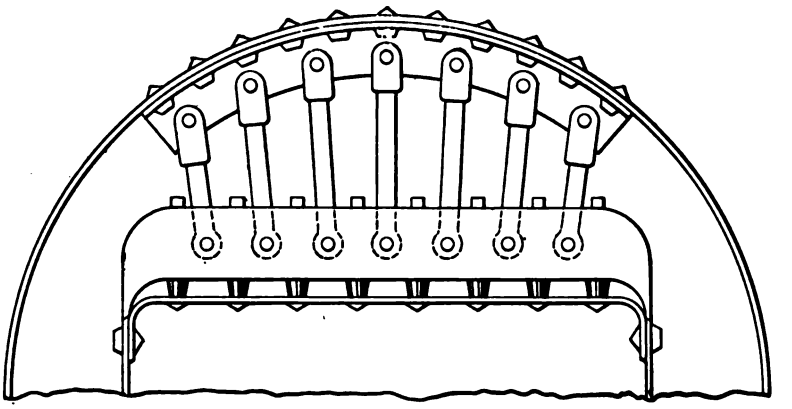


Fig. 38.

side plates. Bolts about 4 inches apart connect the crown sheet to this girder. The girder may be a solid bar, or it may be made up of two flat plates bolted or riveted together, as shown in the figure, the stay bolts being placed between the plates at intervals

of about 4 inches. Either bolts or rivets may be used to keep the plates which form the girder from spreading. Projections are sometimes forged on the bottom of the girder, so that the stay bolts may be screwed up tightly without bending the plate.

The depth of the plates which make up the girder vary from

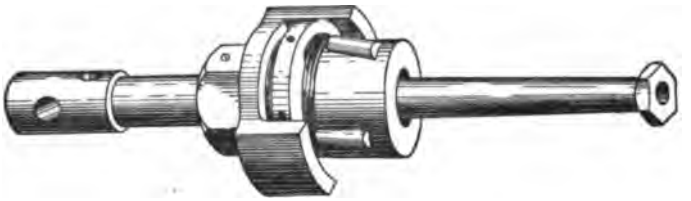


Fig. 39.

4 to 6 inches. They are from $\frac{5}{8}$ to $\frac{3}{4}$ inch in thickness. If bolts $\frac{7}{8}$ inch in diameter are used, the distance between the plates is usually 1 inch, but if larger bolts 1 inch in diameter are used, the distance should be $1\frac{1}{8}$ inches. The ends of the bars which rest upon the side plates should be carefully fitted to make a good bearing, and the area should be sufficient to prevent crushing of

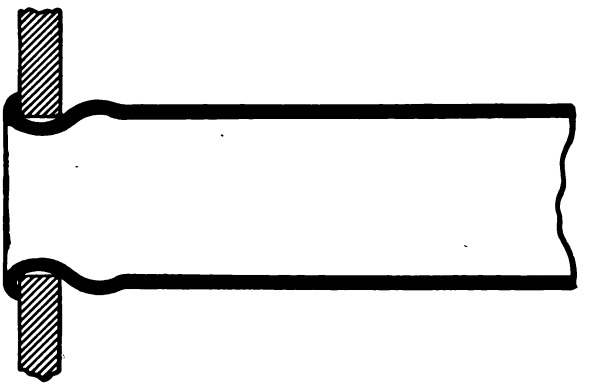


Fig. 40.

the end plates. The distance between the crown sheet and the girder should be at least $1\frac{1}{2}$ inches, so that there will be good circulation and the plates may be readily cleaned.

In some cases the girder is supported from the shell by sling stays, as shown in Fig. 38. The sling stays are connected to the

girder and to an angle iron, or T-iron, which is riveted to the shell. The angle iron stiffens the shell. In designing this form of stay it is usual to make the girder strong enough to support the crown sheet without any sling stays, and these stays are used for additional support.

TUBES.

Boiler tubes are made of steel or wrought iron, but most commonly of charcoal iron and lap welded. In the formation of the lap the plate is upset, then bent around until the thickened edges lap sufficiently. It is then heated successively about 8 inches at a time, and welded over a mandrel, which is a cast-iron

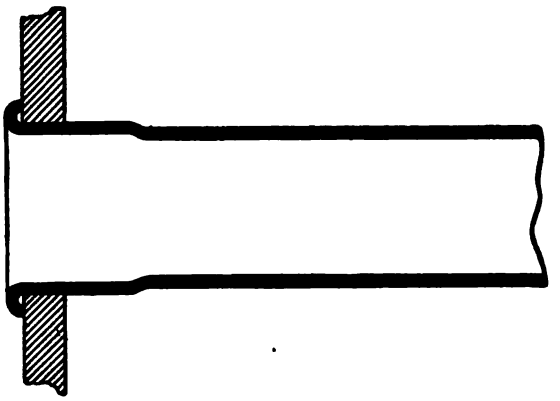


Fig. 41.

arm with a slightly convex top, over which the tube is placed. Tubes are measured by their outside diameters, and are usually true to gauge, so that holes for them may be bored without taking measurements from the tubes themselves.

The holes for the tubes in the tube sheet are usually made in one of two ways. One method is to punch the tube holes the proper size by means of a helical punch. With this punch the metal is cut away by a shearing cut. The holes ought to be punched a little under size, and then reamed out, so that the surface against which the tubes are expanded may be good. The other method is to punch or drill a small hole at the point marking the center of the tube hole. A drill with a post in the center, which fits the small hole, then drills the desired size of hole.

Ordinary tubes are fastened to the end plates by expanding the metal of the tube against the tube plate. This is done by a tool called an expander, of which there are two common forms. One form consists of a steel taper pin and a number of steel segments, held in place by a spring. The outside of the segments

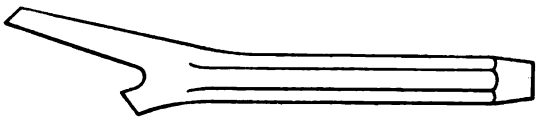


Fig. 42.

have the form to be given to the expanded tube, and the inside is a straight hollow cone, into which the steel taper pin fits. The segments are forced apart by hammering on the steel pin. In order that the metal of the tube may not be injured, the hammering should be done gradually and carefully, and the expander turned frequently. Another form, shown in Fig. 39, has a set of rolls that are forced against the inside of the tube by driving in the taper pin. The pin and rolls rotate as the pin is driven, and the rolls gradually expand the tube against the tube plate.

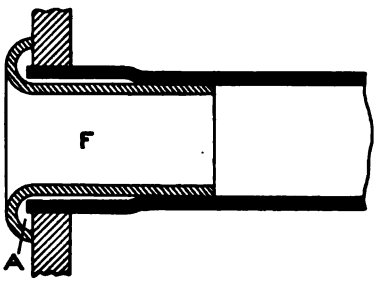


Fig. 43.

Two forms of tube expansion are shown in Figs. 40 and 41. That shown in Fig. 41 is preferable to that in Fig. 40, as the latter bears at the corners only, while the former bears against the entire thickness of the tube sheet.

After the tubes are expanded, the ends are beaded over, as shown in Figs. 40 and 41. This adds to the strength of the connection between the tube and tube sheet. The tool commonly used for this beading is shown in Fig. 42.

Ferrules are often placed in the ends of fire tubes, and serve to protect the ends from the intense heat of the fire. The arrangement is shown in Fig. 43, the ferrule F being placed within the tube for a short distance. The space A is merely an air space.

Stay tubes are not used as extensively at the present time as they were formerly. They were very common at a time when the holding power of expanded tubes had been experimented on but little. It is now apparent from such tests that the holding power of tubes expanded, as shown in Fig. 40, is more than equal to the pressure on the spaces between the tubes of an ordinary tube plate. Stay tubes are simply heavier tubes, with the ends pro-

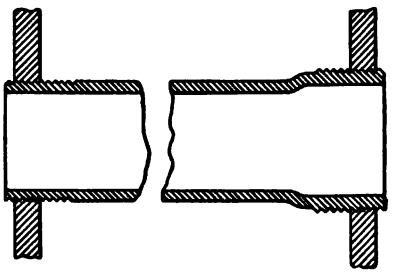


Fig. 44.

jecting beyond the tube sheet and threaded for shallow nuts. The ends of the tubes are frequently upset or thickened, and screwed into the tube sheet as well. This form is shown in Fig. 44.

FURNACE FLUES.

Flues which are subjected to external pressure should always be cylindrical. Fig. 45 shows the section of the Adamson

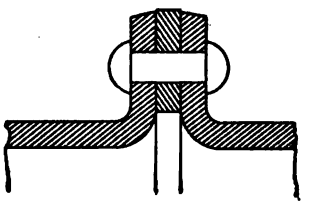


Fig. 45.

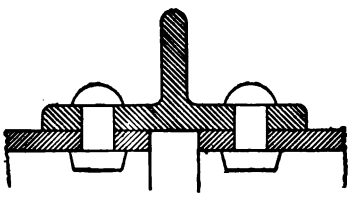


Fig. 46.

flue. This was an improvement over the plain furnace, as it is more elastic and allows expansion; the flanged rings also strengthen and stiffen it against collapse. The methods of building furnaces shown in Figs. 46 and 47 are not considered as good as the Adamson arrangement. Fig. 46 is too rigid, and does not

allow a free expansion and contraction. Fig. 47, on the other hand, permits of such extremely well, but both have the fault of exposing a double thickness of plates and two rows of rivets to the fire.

The corrugated flue shown in Fig. 48 is popular and, furthermore, is excellent. There is freedom for expansion throughout its whole length, thereby reducing the strains on the boiler.

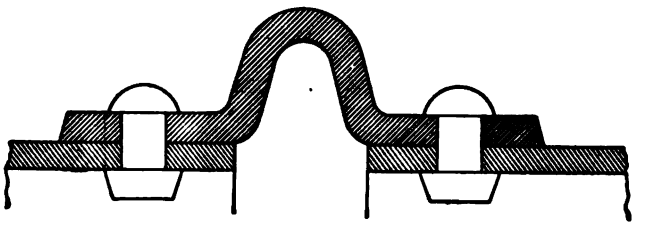


Fig. 47.

The plates should be thick enough to prevent sagging in the middle, the thickness usually varying from $\frac{5}{16}$ inch to $\frac{5}{8}$ inch. Corrugated furnaces are riveted to the rear tube sheet in the return tube boiler of the marine type, the end of the furnace being

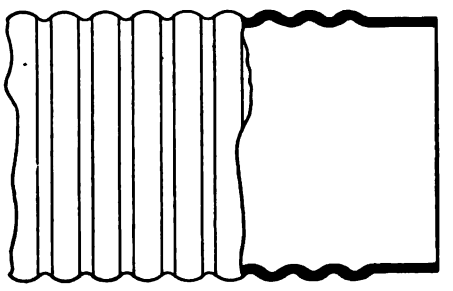


Fig. 48.

flanged at the front; and the head of the boiler is flanged around the opening cut for the furnace, which fits well into the flange.

CALKING.

In order that riveted joints of boilers may be steam and water tight, they generally require calking. This process upsets the metal of the overlapping plate, or burrs down the edge,

forcing it into close contact with the lower plate, and rendering the joint steam tight.

The calking tool is similar to a chisel, the end having a variety of shapes. Fig. 49 shows a round-nosed tool which burrs down the upper plate without cutting the under plate; but it is hard to start, and in calking with such a tool the edge is first started with a sharper round-nosed tool, and then finished with one as indicated in the figure. If a square-end tool is used, as shown in Fig. 50, the under plate is likely to be cut, and the plates between the edge and the rivet be separated. The most common form of calking tool is one similar to the one shown in

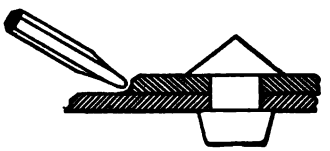


Fig. 49.

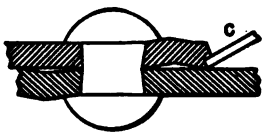


Fig. 50.

Fig. 49, except that the end is flat, with a slight bevel, and not round.

A slight bevel given the plates makes both calking and fullering more easily done. When the calking tool is thin it is sometimes driven by careless workmen into the joint, wedging the plates open. Severe and careless calking is very injurious to boilers. On the inside it often causes grooving and fracture, and the fracture of plates then follows the line of calking rather than the line of rivet holes. A pneumatic calking machine is often used in boiler shops, as it does this work about four times as rapidly as it can be done by hand. It resembles a rock drill in general principles. Air is supplied through a flexible tube, at a pressure of about 70 pounds per square inch. It makes about 1,500 strokes a minute.

BOILER DESIGN.

The rules of boiler design are controlled by practical considerations and theory, and are learned by the designer by practice only. The rules vary from place to place, and from time to time, due to progress in engineering.

The rules, methods and cautions taken up here are general, and with necessary modifications can be applied to all the more common types.

In designing a steam boiler there are several considerations that must be kept in mind. Among the most important are strength, durability, capacity to furnish the required amount of steam, convenience for cleaning, repairing and inspection, simplicity in detail, and economy both of running and first cost.

The kind, or type, to be used depends upon the work to be done, the dryness of the steam, the locality, the available space and preference of the owner. The work to be done is determined by the number and kind of engines, the constancy with which they run and the pressure. In choosing a boiler for any locality, the purity of the water, the kind of fuel and the laws which govern inspection and allowable working stress must be considered. The available space greatly influences the type and sometimes prevents choice. For instance, locomotive and marine boilers must be put in a small space. For land boilers if the floor area is limited, but there is ample height, some type of vertical boiler must be chosen.

HORSE POWER.

The unit of horse-power as decided by the American Society of Mechanical Engineers is equal to 33,305 B. T. U. From the standard steam tables in treatises on Thermo-dynamics we find that 966 B. T. U. are required to evaporate one pound of water from and at 212° F. Therefore 1 H. P. is equal to the evaporation of $33{,}305 \div 966 = 34\frac{1}{2}$ pounds of water from and at 212° F. This is also equal to the evaporation of 30 pounds of water, at 100° F. into steam at 70 pounds gauge pressure.

The first thing to do is to chose the type of boiler we are to use. Then we find how many pounds of steam are to be supplied per hour; this is found by multiplying the desired horse-power by

34½ or multiplying the horse-power of the engine or engines by the steam consumption per horse-power per hour. This is known approximately for every type of engine.

GENERAL REQUIREMENTS.

When we know these facts we design our boiler so as to have:

1. Sufficient area of grate to burn the required amount of fuel under the given draft.
2. Enough heating surface to absorb the heat of combustion.
3. Combustion chamber and flue area large enough to completely burn and carry off the products of combustion.
4. Water space sufficiently large so that a sudden demand will not cause too great a variation in water level.
5. Surface of water large compared to volume, in order that steam may be rapidly disengaged.
6. Steam space large enough to supply an irregular demand without causing a great change of pressure.
7. Steam outlet large enough to supply steam to the engine without wire-drawing.

If the outlet is not sufficiently large to supply plenty of steam, the demand will be greater than the supply and the steam will be throttled or wire-drawn, that is, it will lose some pressure.

For all common types of boilers, the proportions between the above requisites have been determined by experiment and mathematics. These relations, with simple calculations and good judgment on the part of the designer, are all that are needed for this work.

AREA OF GRATE.

A square foot of grate area will burn different weights of fuel in a given time, according to the nature of the draft. If the boiler can be made of any size, as is the case with many land boilers, a slow rate of combustion with natural draft is used, as it is the most economical. The length of the grate is limited by the distance to which a fireman can throw coal accurately. Usually 6 or 7 feet is the limit. In locomotive, torpedo boat and in some vertical land boilers, the size of grate is limited; in order

to get the necessary work from the boiler, forceddraft is used and the rate of combustion increases to over 100 pounds per square foot per hour. In Lancashire boilers, with two internal flues, the breadth is limited. The rate of combustion is stated in pounds per square foot of grate area per hour, and varies with the type of boiler and the draft. The following table gives the rates of combustion.

CHIMNEY DRAFT.

Cornish boilers, slow rate	4—6 lbs. per sq. ft. per hour.
Cornish boilers, ordinary rate	10—15 lbs. per sq. ft. per hour.
Factory boilers, ordinary rate	12—18 lbs. per sq. ft. per hour.
Anthracite coal, quick rate	15—20 lbs. per sq. ft. per hour.
Bituminous coal, quick rate	20—30 lbs. per sq. ft. per hour.
Marine boilers, ordinary rate,	15—25 lbs. per sq. ft. per hour.
Water tube boilers	10—25 lbs. per sq. ft. per hour.

FORCED DRAFT.

Marine boilers,	60—130 lbs. per sq. ft. per hour.
Locomotive boilers,	40—120 lbs. per sq. ft. per hour.

The evaporation per square foot of grate surface depends upon the type, the rate of combustion, condition of boiler and care in firing. The highest rate is obtained with slow rate of combustion, care and skill in firing, and clean plates and tubes. The table gives the equivalent evaporation per pound of coal for several types.

Plain cylindrical	5— 8 pounds.
Vertical	7—10 pounds.
Cornish	6—11 pounds.
Lancashire	6½—12 pounds.
Galloway	9—12½ pounds.
Multitubular	8—12 pounds.
Water tube	6 —12 pounds.
Marine return tube	7—12 pounds.
Locomotive	6—12 pounds.

Experiment shows that an increase in the amount of coal burned per square foot of grate per hour gives an increase in the

amount of water evaporated; but a decrease in amount evaporated per pound of fuel, or a decrease in economy.

To find the area of grate for a boiler. Let G = area of grate in square feet, R = rate of combustion in pounds per square foot per hour, E = evaporation per pound of coal.

$$\text{Then } G = \frac{\text{Pounds of water evaporated}}{E \times R}$$

Let us take an example. Suppose we have an externally fired multitubular boiler; assume the rate of combustion to be 12 pounds, and that our type of boiler will evaporate 9 pounds of water per pound of coal. How large must the grate be, if 2400 pounds of water are evaporated per hour?

$$G = \frac{2400}{E \times R} = \frac{2400}{9 \times 12} = 22.2 \text{ square feet.}$$

Then 22.2 square feet of grate surface are necessary. In this case the grate probably would be made 6 feet by 4 feet or 24 square feet.

TUBES.

On account of the small number of successful experiments concerning flues and chimneys, it is usual to proportion tubes, flues and chimneys, by comparison with those that have given good results. If the tubes are too large the hot gases in the centre pass up the chimney at high temperature. Now we will find the number of tubes. Let A = total area in square feet through which the smoke passes, that is, the combined internal area of all the tubes. The total area of the tubes, A, is usually made $\frac{1}{6}$ to $\frac{1}{8}$ the area of the grate. If we design our boiler to have the ratio 1 : 8 we probably will have enough area. Let us assume our tubes to be 3 inches in diameter and 16 feet long. From the table, on page 29, of lap welded boiler tubes we find that the internal area of a 3 inch tube is 6.08 square inches, the internal circumference is 8.74 inches, external circumference is 9.42 inches, and the external area is 7.07 square inches. As $\frac{1}{8}$ of our grate surface is $\frac{24}{8}$ or 3 square feet, or 432 square inches, the number of tubes will be $432 \div 6.08 = 71$.

LAP WELDED BOILER TUBES.

External Diameter. Inches.	Internal Diameter. Inches.	Thickness. Inches.	Internal Circumference. Inches.	External Circumference. Inches.	Internal Area. Square Inches.	External Area. Square Inches.	Length of tube per sq. ft. inside. Feet.	Length of tube per sq. ft. outside. Feet.	Weight per foot. Lbs.
1	.856	.072	2.689	3.142	.575	.785	4.460	3.819	.708
1¼	1.106	.072	3.474	3.927	.960	1.227	3.455	3.056	.900
1½	1.334	.083	4.191	4.712	1.396	1.767	2.863	2.547	1.25
1¾	1.560	.095	4.901	5.498	1.911	2.405	2.448	2.183	1.665
2	1.804	.098	5.667	6.283	2.556	3.142	2.118	1.909	1.981
2¼	2.054	.098	6.484	7.069	3.314	3.976	1.850	1.698	2.238
2½	2.283	.109	7.172	7.854	4.094	4.909	1.673	1.528	2.755
2¾	2.533	.109	7.957	8.639	5.039	5.940	1.508	1.390	3.045
3	2.783	.109	8.743	9.425	6.083	7.069	1.373	1.273	3.333
3¼	3.012	.119	9.462	10.210	7.125	8.296	1.268	1.175	3.958
3½	3.262	.119	10.248	10.995	8.357	9.621	1.171	1.091	4.272
3¾	3.512	.119	11.033	11.781	9.687	11.045	1.088	1.018	4.590
4	3.741	.130	11.753	12.566	10.992	12.566	1.023	.955	5.32
4½	4.241	.130	13.323	14.137	14.126	15.904	.901	.849	6.01
5	4.720	.140	14.818	15.708	17.497	19.635	.809	.764	7.226
6	5.699	.151	17.904	18.849	25.509	28.274	.670	.637	9.346
8	7.636	.182	23.989	25.132	45.795	50.265	.500	.478	15.109
10	9.573	.214	30.074	31.416	71.975	78.540	.399	.382	22.190
12	11.542	.229	36.260	37.699	103.749	113.097	.330	.318	28.516
16	15.458	.271	48.562	50.265	187.667	201.062	.247	.238	45.200
20	19.360	.320	60.821	62.832	294.373	314.159	.197	.190	66.765

STEAM SPACE.

The steam space is frequently designed as some fraction of the volume of the shell, usually about $\frac{1}{3}$. A better way is to design it from the steam consumption of the engine. Suppose the engine uses 30 pounds of steam at 75 pounds pressure per H. P. per hour. The absolute pressure then is 90 pounds (nearly) and the specific volume at that pressure is 4.85 (from steam tables). As steam is being generated at an approximately constant rate, the supply kept on hand need not be great. If the surface for the disengagement of steam is sufficient, the ratio of the steam space to the volume of the cylinder is from 50 : 1 to 150 : 1 depending upon the speed of the engine. Experiment shows that if the steam space is equal to the volume of steam consumed by the engine in 20 seconds, it is sufficient. If the space is only equal to the steam used in 12 seconds, there may be a considerable quantity of water carried over with the steam. If the engine is slow speed, that is less than 60 revolutions per minute, the steam space should be larger.

The volume of the steam space per H. P. will be the number of pounds of steam used per H. P. in 20 seconds, multiplied by its specific volume, or $\frac{30 \times 4.85 \times 20}{60 \times 60} = .81$ cubic feet (nearly) per H. P.; and if the engine is of 75 H. P. our steam space will be $.81 \times 75 = 60.75$ cubic feet.

TUBE SPACE.

The space occupied by the tubes is equal to their volumes. The volume of one tube is its external area multiplied by the length in inches. The total volume, in cubic inches, is the above result multiplied by the number of tubes. This is reduced to cubic feet by dividing by 1728. The space occupied by the tubes will be

$$\frac{71 \times 7.07 \times 16 \times 12}{1728} = 55.77 \text{ cubic feet.}$$

WATER SPACE.

Then if we assume our steam space to be $\frac{1}{3}$ the volume of the

available space in the shell, the water space will be twice the steam space, or $2 \times 60.75 = 121.5$ cubic feet.

DIMENSIONS OF BOILER.

The volume of the boiler will be:

Steam space	$.81 \times 75 =$	60.75 cubic feet.
Tube space	$\frac{71 \times 7.07 \times 16 \times 12}{1728} =$	55.77 cubic feet.
Water space	$.81 \times 75 \times 2 =$	121.5 cubic feet.
Total space		238.02 cubic feet.

Since the tubes are 16 feet long the area of the end will be

$$\frac{238.02}{16} = 14.87 \text{ square feet.}$$

This area gives a diameter of about $4\frac{1}{3}$ feet or 52 inches. We will make the boiler $4\frac{1}{2}$ feet or 54 inches in diameter. Then the boiler will be 16 feet long and 54 inches in diameter; with 71 tubes 3 inches in diameter. For moderate power, a common rule is to make the length about $3\frac{1}{2}$ times the diameter; by this rule our boiler is 3.55 times the diameter.

HEATING SURFACE.

The portion of a boiler that is exposed to the flames and hot gases is called the heating surface. This is made up of the portions of the shell below the brickwork, the exposed ends, and the internal surface of the tubes. If the boiler is of the water tube type, the exterior surface of the tubes is taken in place of the interior surface.

If our boiler is an ordinary multitubular boiler we can assume the heating surface to be the total inside area of the tubes plus one-half the area of the shell. Then:

Heating surface of tubes	$\frac{8.74 \times 71 \times 16}{12} =$	827.38 square feet.
Heating surface of shell	$\frac{14.137 \times 16}{2} =$	113.10 square feet.
		940.48 square feet.

The ratio of heating surface to grate surface will be

$$\frac{940.48}{24} = 39.2 \text{ or about } 39.$$

As this ratio is high enough we will not alter our figures. If the ratio had been too low we could have added more tubes and found a new boiler diameter. The heating surface should not be less than 1 square yard or 9 square feet per horse-power. So $940.48 \div 75 = 12.54$ or our boiler has 12.54 square feet of heating surface per horse-power. This is of course abundantly sufficient.

The capacity of heating surface to transmit heat to water depends upon conductivity, position of surface and temperature of furnace. In designing it is safe to follow proportions of heating surface to grate area in the various types, which experience has shown to give the best results. The following are the proportions for a few types.

Kind of Boiler.	Ratio of Heating Surface to Grate Surface.
Marine, Return tube,	25—38
Lancashire boiler,	26—33
Cornish,	27—32
Horizontal, internally fired,	40—50
Water tube,	34—65
Locomotive boiler (forced draft),	30—34
Marine,	28—32

RATIO OF GRATE SURFACE TO HORSE-POWER.

The ratio of grate surface to horse-power varies with the type, as is shown below.

Kind of Boiler.	Ratio.
Plain cylindrical,	.5 to .7
Multitubular,	.4 to .6
Vertical,	.6 to .7
Water tube,	.3
Lancashire,	.1 to .165
Marine return tube,	.12
Locomotives,	.02 to .06

Makers of boilers sometimes estimate the H. P. by the heating surface. That is the horse-power is a fraction of the heating surface. The ratio of heating surface to H. P. for several types is as follows:

Plain cylindrical,	6 —10
Multitubular,	14 —18
Vertical,	15 —20
Water tube,	10 —12
Marine return tube,	3.25— 4
Lancashire,	2.75— 4.25
Locomotive,	1 — 2

It is evident that some portions of the heating surface of a boiler have greater efficiencies than others. For instance, more heat will pass through the crown sheet as it is nearer the fire than through the last few feet of the tubes. Taking the efficiency of the crown sheet as 1, an estimate of the percentage of the other parts of a boiler is as follows:

Crown of furnace in flue,	.95
Plates of cylindrical boiler over furnace,	.90
Fire box tube plate of locomotive boiler,	.80
Water tube surface facing fire,	.70
Vertical side of fire box,	.50

If a cylindrical multitubular boiler is divided into equal sections, the section nearest the fire will evaporate more water than the one at the other end, as the gases have a higher temperature at the first section. Suppose we divide the boiler into six sections of equal length, and call the total evaporation 100 per cent. Then the per cent of evaporation per section will be approximately as follows:

Section	1	2	3	4	5	6
Evaporation	47	23	14	8	5	3

If the length of a boiler is increased another section, the evaporation will be increased a little but at the same time the radiating surface is increased. In case the addition of a section for evaporation causes a loss by radiation nearly equal to the gain in evaporation, it is not economical to add the section on account of the extra cost of the boiler. If forced draft or an increase of air of dilution is used, the boiler should be made longer to avoid waste. The air of dilution is the amount of air above that which is necessary to burn the coal.

WATER LEVEL.

If the steam space in a multitubular boiler is known the water level can be found, for the section of the steam space is a segment of a circle. In the above boiler the required steam space is 60.75 cubic feet; hence the segmental area is 60.75÷16 or 3.8 square feet, or 547.2 square inches. The height of this segment is 15.55 inches. This height is found either by calculation or from a table of segments. Then the mean water level is 15.55 inches from the top portion of the shell. The variation of water level in a boiler of this type and size should not exceed 6 inches.

END PLATE.

The end plate or tube sheet is usually made $\frac{1}{16}$ or $\frac{1}{8}$ inch thicker than the shell plates. This is done for additional stiffness, and increase of strength; the plate being weakened by drilling the holes for the ends of the tubes.

The tubes should be arranged in vertical and horizontal rows, if possible, in order that the rising bubbles of steam may not be hindered. To get good circulation the horizontal spaces should be a little greater than the vertical, and a central circulating space should be provided, if the necessary number of tubes can be put in without using the entire space. The tubes should be from $\frac{3}{4}$ to 1 inch apart, and to prevent burning of the tubes, the top row at least 3 inches below the water level, and the bottom tubes 6 inches from the shell. At this point, a drawing of the end plate should be made, to show the arrangement of tubes, etc. If it is impossible to put in the required number of tubes, without raising the water level, the diameter of the boiler must be increased. If we wish to increase the heating surface without increasing the diameter we can use smaller tubes or make the boiler a little longer.

STRENGTH OF BOILERS.

According to Pascal's Law, liquids and gases exert pressure equally in all directions. Steam in a boiler exerts the same pressure on all portions of the shell. As the pressure inside a boiler is considerably greater than that outside (the atmospheric pres-

sure), there is a tendency to burst the shell. This tendency is resisted by the plates of the boiler.

A sphere is the strongest form to resist pressure, for since pressure is equal in all directions, there is a tendency towards enlarging the sphere and not to rupture. But a sphere has the smallest area for a given volume and, as a large heating surface is desirable, and on account of mechanical difficulties, a spherical boiler is never used. The boiler is made cylindrical to obtain greater heating surface and the loss in strength is made up by staying.

In the consideration of the strength of cylinders it is usual to divide the rupturing strains into two classes; those which tend to rupture the cylinder longitudinally and those which tend to rupture it circumferentially or transversely.

Let us examine them separately. The tendency to cause longitudinal rupture or to rend the cylinder in lines parallel with the axis, may be considered as the pressure exerted on a semi-circumference, and tending to rupture the cylinder in a plane through the diameter. Since pressure acts equally in all directions, the whole amount exerted on a semi-circumference is not exerted directly upwards and downwards. But all these forces may be resolved into their vertical and horizontal components. If we take the plane as horizontal, it is evident that the horizontal components have no tensional effect at the points of rupture. By taking the vertical components at an infinite number of points it can be proved that their sum is equal to the full pressure exerted on a rectangular plane equal to the projection of the cylindrical surface. In this case the projection is the plane through the diameter and has an area equal to the product of the length of the cylinder multiplied by the diameter of the cylinder. Then the force tending to rupture would be the pressure per square inch multiplied by the area. Let p = pressure in pounds per square inch, D = diameter of boiler, t = thickness of plate, L = length of boiler, S = tensile strength, E = efficiency of joint, and f = factor of safety. The force tending to rupture longitudinally will be, pLD. The strength of the cylinder to resist this rupturing force is represented by the tensile strength of the material multiplied by the areas of sections of metal. Or expressed

algebraically is 2tLS. When rupture is about to take place the rupturing force and the strength are equal, or

$$pDL = 2tLS \quad \text{or} \quad pD = 2tS$$

from which $p = \frac{2\,t\,S}{D}$ and $t = \frac{pD}{2S}$ which are the formulas for pressure and thickness and for longitudinal strength.

The extra pressure due to increased length is balanced by the increase of metal as is shown by the elimination of the factor L of the equation.

The tendency to rupture circumferentially is evidently represented by the area of the end or $\frac{\pi D^2}{4}$ multiplied by the pressure per square inch. The strength to resist this force is the area of metal to be ruptured multiplied by the tensile strength or π DtS

$$\frac{\pi D^2}{4} \times p = \pi DtS$$

$$Dp = 4tS$$

By comparing these two formulas we see that with the same internal pressure, diameter and thickness of shell, a cylindrical boiler is twice as strong transversely as it is longitudinally, hence the greatest tendency to rupture is usually, but not always, along the longitudinal seams.

Therefore, in designing the thickness of shell we use the formula for longitudinal rupture,

$$pD = 2tS \text{ or } t = \frac{pD}{2S}$$

or, inserting the factors for efficiency of joint and factor of safety,

$$pD = \frac{2tSE}{f}$$

$$\text{For allowable pressure} \quad p = \frac{2tSE}{fD}$$

$$\text{For thickness of shell} \quad t = \frac{fDp}{2SE}$$

Now let us find the thickness of the boiler that we are designing. Suppose after testing our material we find that its ultimate tensile strength is 54,000 pounds per square inch. In this case 6 will be sufficiently large for a factor of safety. This factor can

be reduced if the efficiency of the joint is large. Let us assume that our joint has an efficiency of 70%. This is merely a supposition because we have not yet constructed the joint; but we assume a factor in order to find a trial thickness.

$$\text{Then } t = \frac{fDp}{2SE} = \frac{6 \times 54 \times 75}{2 \times 54000 \times .7} = .32 \text{ or about } \tfrac{5}{16} \text{ inches.}$$

RIVETED JOINTS.

The best knowledge of the strength and proportions of riveted joints can be obtained by tests of full sized pieces. Let us consider the strength and efficiency mathematically. Riveted joints may fail in several ways. 1. By shearing the rivets. 2. By tearing the plate at the reduced section between the rivets. 3. By crushing the plate or rivets where they are in contact. 4. By cracking the plate between the rivet hole and the edge of the plate. As the lap in practice can always be made sufficiently wide a joint need never fail in this last way.

The following mathematical consideration of riveted joints as far as the star * may be omitted if the student desires.

As all stresses may be resolved into the three kinds, tensile, compressive and shearing, we will investigate for these stresses. Let P = the tensile stress transmitted from one plate to the other by a single rivet, t = the thickness of the plate, d = the diameter of the rivet, p = the pitch, and S_t, S_s and S_c the unit stresses in tension, shear and compression respectively produced by P on the plates and rivets. Therefore the tension on the plate, P will be equal to the area of the metal between the rivets multiplied by its unit tensile stress, or

$$P = t\,(p - d)\,S_t$$

For shear, P will equal the area of the rivet multiplied by the unit shearing stress, or

$$P = \tfrac{1}{4}\,\pi\,d^2 S_s$$

For compression, the stress is supposed to be equivalent to a stress uniformly distributed over the projection of the cylindrical surface on a plane through the axis of the rivet. Then P will be equal to the area of the projection multiplied by the unit compressive stress, or

$$P = tdS_c$$

The above formulas are for single riveted lap joints. If another row of rivets is used the plates should have a wider lap. Let p = the pitch in one row; then the stress will be distributed over two rivets.

The three formulas in this case will be;

$$P = t\,(p - d)\,S_t$$
$$P = 2 \times \tfrac{1}{4}\,\pi\,d^2 S_s$$
$$P = 2\,tdS_c$$

For single riveted butt joint, the sheer comes on two rivets; this is called double shear. The above formulas become

$$P = t\,(p - d)\,S_t$$
$$P = 2 \times \tfrac{1}{4}\,\pi\,d^2 S_s$$
$$P = tdS_c$$

The efficiency of a joint is the ratio of its allowable stress to the allowable stress of the uncut plate. The allowable stress of the plate is represented by the formula ptS_t.

Then the efficiency for tension is, $E = \dfrac{t\,(p - d)\,S_t}{ptS_t} = \dfrac{p - d}{p}$

For shear, $E = \dfrac{\frac{1}{4}\,\pi\,d^2 S_s}{ptS_t}$ or $\dfrac{\frac{1}{4}\,\pi\,d^2 S_s c}{ptS_t}$

For compression, $E = \dfrac{tdS_c}{ptS_t}$ or $\dfrac{dS_c}{pS_t}$ or $\dfrac{dS_c a}{pS_t}$

In the above formulas, a = the number of rivets in the width p, and c = the number of rivet sections in the same space. The smallest value of E is to be taken as the efficiency of the joint.

In designing, we try to get a joint in which all parts will have equal strength or the resistance of the plate to tension will equal the resistance of the rivets to shearing and each will equal the resistance of the rivet to compression or crushing. This will be the case if the three efficiencies are equal.

Solving for d in the second and third we get

$$\frac{\frac{1}{4}\,\pi\,d^2 S_s c}{ptS_t} = \frac{dS_c a}{pS_t} \text{ or } d = \frac{4\,aS_c t}{\pi\,cS_s}$$

If we know t we can find d from the above equation.

To find the pitch we make the first equation equal to the third, or the formula for tension equal that of compression, and solve for p

$$\frac{p-d}{p}=\frac{dS_c a}{pS_t},\ p=d\left(\frac{S_c a}{S_t}+1\right)$$

substituting the value for d, obtained above,

$$p=\frac{4aS_c t}{\pi cS_s}\left(\frac{S_c a}{S_t}+1\right).$$

To get the formula for efficiency we insert these values for d and p, in any of the formulas for efficiency already obtained. For instance:

$$E=\frac{p-d}{p}=\frac{\frac{4aS_c t}{\pi cS_s}\left(\frac{S_c a}{S_t}+1\right)-\frac{4aS_c t}{\pi cS_s}}{\frac{4aS_c t}{\pi cS_s}\left(\frac{S_c a}{S_t}+1\right)}$$

$$E=\frac{1}{1+\frac{S_t}{aS_c}}$$

A good joint can be designed without these formulas (in fact they serve as a guide only), if attention is paid to the rules deduced from tests and conforming to good practice by experienced engineers and boiler makers. In designing a riveted joint, good practice favors the following:

The pitch of rivets, for single riveting, should be about $2\frac{1}{2}$ times the diameter of the rivets and for double riveting about $3\frac{1}{4}$ times the diameter.

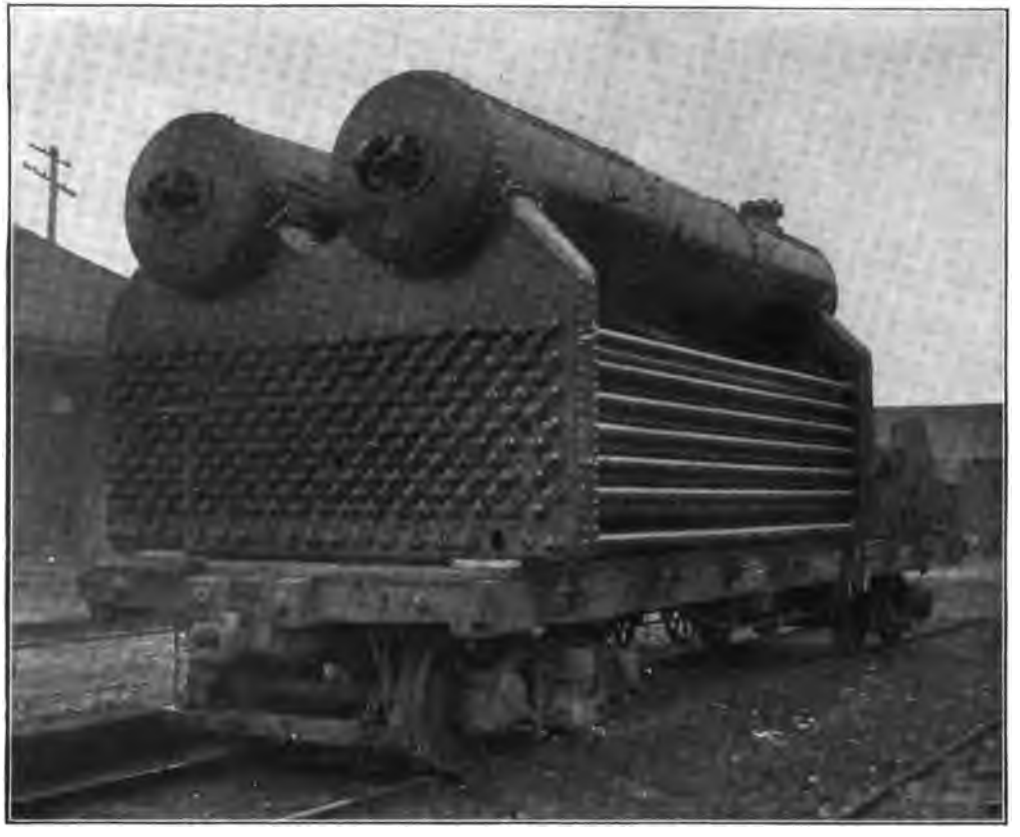
The pitch near a calked edge must not be too great for proper calking.

Rivets must not be too near together.

The lap, or the distance from the centre of the rivet to the edge of the over-lapping plate should be at least $1\frac{1}{2}$ times the diameter of the rivet.

The diameter of the rivet is usually nearly twice the thickness of the plate and should never be less than the thickness of the plate.

The riveted seam must contain a whole number of rivets, and similar seams should have the same pitch.



500 Horse-power Heine Boiler Ready for Transportation.

The distance between rows, for double riveting, is about twice the diameter of the rivets.

In double butt riveting the rivets in double shear have $1\frac{8}{4}$ times the single section instead of 2.

The thickness of double butt straps should not be less than $\frac{5}{8}$ the thickness of the plate (each); single butt straps not less than $\frac{9}{8}$.

No one set of rules can be laid down for the best pitch of rivets for all circumstances of pressure, quality of plates, etc. The following table of proportions of riveted joints gives results for average practice in boilers of up to about 150 pounds pressure.

TABLE OF LAP JOINTS.

Thickness of Plate. Inches.	Diameter of Rivet. Inches.	Diameter of Hole. Inches.	Pitch. Inches.		Efficiency.	
			Single Riveted.	Double Riveted.	Single Riveted.	Double Riveted.
$\frac{1}{4}$	$\frac{5}{8}$	$\frac{11}{16}$	2	3	.66	.77
$\frac{5}{16}$	$\frac{11}{16}$	$\frac{3}{4}$	$2\frac{1}{16}$	$3\frac{1}{8}$	.64	.76
$\frac{3}{8}$	$\frac{3}{4}$	$\frac{13}{16}$	$2\frac{1}{8}$	$3\frac{1}{4}$	.62	.75
$\frac{7}{16}$	$\frac{13}{16}$	$\frac{7}{8}$	$2\frac{3}{16}$	$3\frac{3}{8}$	.60	.74
$\frac{1}{2}$	$\frac{7}{8}$	$\frac{15}{16}$	$2\frac{1}{4}$	$3\frac{1}{2}$	.58	.73

As the stress on the transverse section is one-half that on a longitudinal section, a single or double lap joint is sufficient for any ring seam. For externally fired multitubular boilers with shell plates less than $\frac{1}{2}$ inch thick, single riveted ring seams are used. For our boiler, the plates being $\frac{5}{16}$ inch thick, we will use rivets $\frac{11}{16}$ inch in diameter, as this agrees with good practice. From the table, the pitch for a $\frac{11}{16}$ inch rivet for single riveted lap joint is $2\frac{1}{16}$. Then as our ring seam is $3.1416 \times 54 = 169.65$ inches and pitch $2\frac{1}{16}$ inches we will have 82 + rivets. But as we must have a whole number of rivets we will alter the pitch slightly and use 82 rivets with a pitch of 2.069 inches. The result depends, in each case, upon the kind of joint used in longitudinal seams. This merely shows the general method. The lap will be $\frac{11}{16} \times \frac{3}{2} = 1\frac{1}{32}$ inch.

For the longitudinal seams we will use double butt joints with single riveting. The thickness of the butt straps will be $\frac{5}{16} \times \frac{5}{8} = .20$ (nearly). To be on the safe side we will make the butt straps $\frac{1}{4}$ inch thick. The pitch for double butt joints is usually about 4 times the diameter of rivets, so in this case it will be $4 \times 1\frac{1}{16} = 2\frac{3}{4}$ inches. We will use the same amount of lap for this joint as for the lap joint, that is $1\frac{1}{32}$ inches.

SECTIONS.

The boiler is made up of rings or sections. The length of sections is often made equal, for convenience in ordering and cutting plates. The length is limited by the width of plate obtainable and the size of the riveting machine. This boiler being 16 feet long would probably be made in three sections, but the lengths should be so adjusted as not to bring the ring seam over the hottest part of the fire.

FLUES.

The internal pressure at which the boiler shell will rupture can be calculated; but the external pressure which will collapse a flue can be determined only by experiment. External pressure tends to increase any imperfection of shape. For instance, if a flue is slightly oval, the external pressure tends to make it more flat. The strongest form to resist external pressure is evidently the circle. When considering the strength of flues length is very important.

If a lap joint is used the flue will not be a true cylinder, for this reason welded or butt joints are preferable.

Fairbain gives the formula, $P = \frac{806{,}000\ t^{2.19}}{ld}$ for calculating the collapsing pressure of flues, l = length of flue in feet, d = diameter in inches, t = the thickness in inches, P = pressure per square inch. The exponent of t is often taken as 2 instead of 2.19 for convenience. This formula is empirical and was prepared from his experiments.

Hutton gives, $P = \frac{C\ t^2}{d\sqrt{L}}$. In which C is a constant, which is 600 for wrought iron and 660 for mild steel, L = length in

inches, d = external diameter in inches, and t = thickness in *thirty-seconds* of an inch. Results by Hutton's formula agree more nearly to those by experiment than do Fairbain's.

If the flues are oval, d in the above formula = the major axis.

Flues are strengthened by putting in hoops at stated distances. These hoops are made of T iron or angle iron.

TUBES.

The materials for tubes are iron and steel. The tubes must be tough to resist cutting by cinders. If iron is used it should have a tensile strength of at least 45,000 pounds per square inch, with an elongation of 15 to 20 per cent. If steel, the elongation should not be less than 26 per cent, when tested before being rolled. If the steel welds well there need not be any limit to its tensile strength. The ends of tubes should be annealed after manufacture. The thickness of tubes is always greater than that required to prevent collapsing, in order to weld and expand in the tube sheet. It is often desirable to use part of the tubes as stays; for this purpose the tubes are made thick enough to take a shallow nut outside the tube plate.

STAYING.

As large a portion as possible of the shell of a boiler is made cylindrical, for in this form plates can be made sufficiently strong without the aid of stays or braces. But all flat surfaces must be stayed; not only to prevent rupture, but also to provide against distortion and grooving. The theoretical investigation of the strength of flat surfaces, can be worked out only with higher mathematics. From the formula deduced, the solid end plate would have to be about 2 inches thick for a boiler only 3 feet in diameter with plates $\frac{3}{8}$ inch thick. It is evident that the flat ends if of ordinary thickness must be strengthened by stays or braces. The calculation of stresses in a flat plate, supported by stays, can be calculated only when the supported points are in rows thus dividing the surface into equal squares. Even when the stays are not to be placed in rows forming squares, it is well to make the calculation for a standard.

The equation for finding the area supported by a stay rod is,

$$a^2 = \frac{9t^2S}{2p}$$

in which a^2 = the area supported, t = the thickness of the end plate, S = the allowable stress on the area of the rod, and p = the working steam pressure. Let us find the area supported in our multitubular boiler. In order to provide for future corrosion we will use a factor of safety of 12 and assume, in the absence of exact knowledge, the ultimate breaking strength of the rod to be 60,000 pounds per square inch. It is usual to make the diameter of the rods one to two inches, so we will make ours $1\frac{1}{2}$ inches in diameter with an area of 1.767 square inches. Then the stress per rod is $5{,}000 \times 1.767 = 8{,}835$ pounds.

$$a^2 = \frac{9t^2S}{2p} = \frac{9 \times \frac{1}{4} \times 8{,}835}{2 \times 75} = 132.5 \text{ square inches.}$$

Then as the rod supports 132.5 square inches and the segment of the steam space is 547.2 square inches, the number of rods will be $\frac{547.2}{132.5} = 4$.

The same formula will apply in finding the number of short screw stay bolts of the fire box.

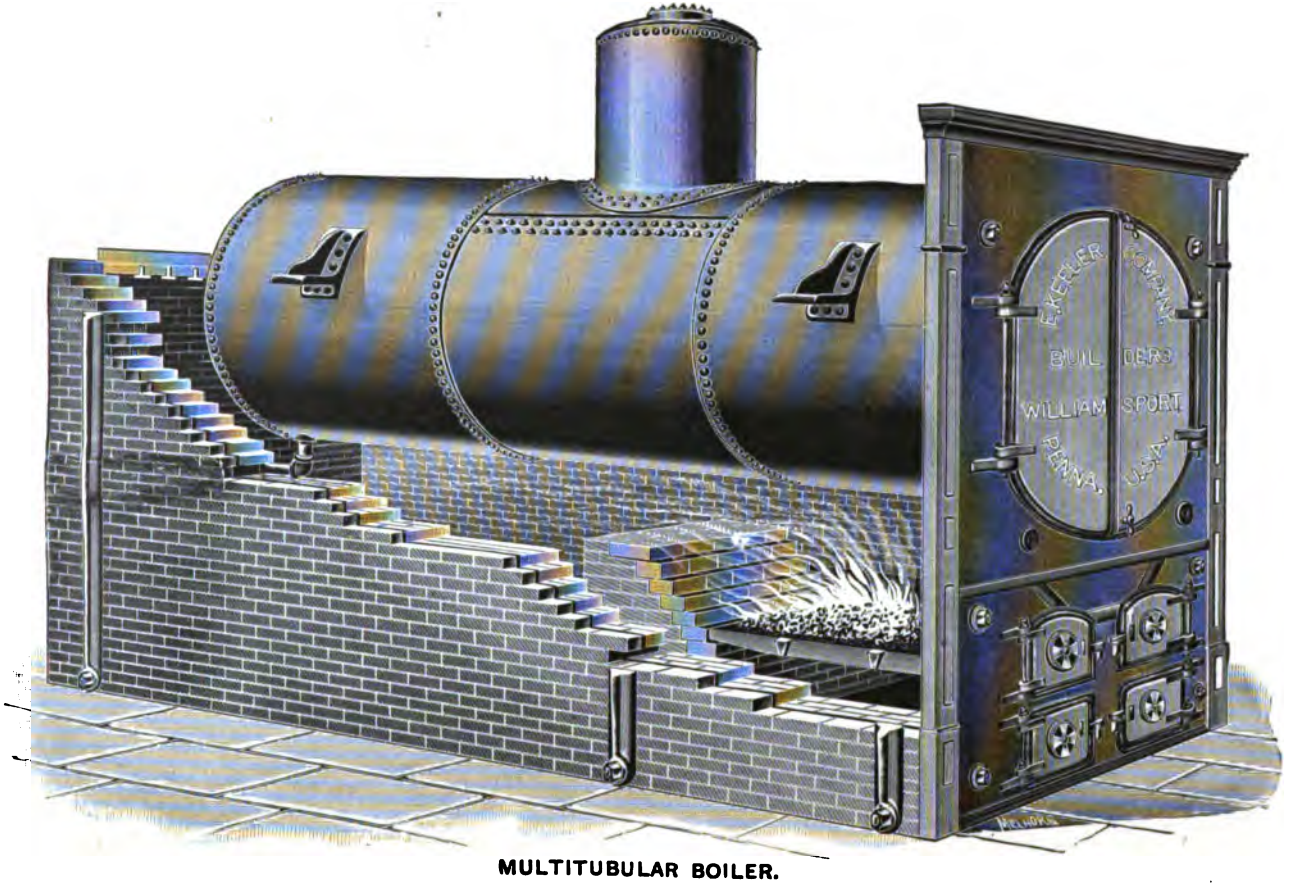
Suppose we wish to use a diagonal or crow foot stay, making an angle of 20° with the shell. If the rod is 1 inch in diameter and the stress is limited to 7,000 pounds, then it will carry a pull of $.7854 \times 7{,}000 = 5497.8$ pounds, and since it makes an angle of 20°, the pull perpendicular to the head will be $5497.8 \times \cos. 20° = 5497.8 \times .9397^* = 5{,}166$ pounds. If the end is fastened by two rivets or bolts each will carry 2,583 pounds. If each rivet or bolt supports a square with a side equal to a, then $5{,}166 = 75\ a^2$

$$a^2 = \frac{5166}{75} = 68.9 \text{ square inches (nearly).}$$

*Note. Taken from a table of cosines.

UPTAKE.

The area of the uptake, like the area of the tubes, is made about $\frac{1}{7}$ to $\frac{1}{8}$ of the area of the grate. We find that $\frac{1}{8}$ of the grate surface is 432 square inches. If we make the uptake 12 inches deep measured with the length of the boiler, it will be $432 \div 12$



MULTITUBULAR BOILER.

= 36 inches wide. The opening of the shell at the front end will be 12 inches deep and the plate cut down until it is 36 inches wide.

MANHOLES.

The manhole and handhole should be strong enough and stiff enough to sustain the stresses due to the direct steam pressure and from the stresses of the plates. The calculation of the strength of the manhole ring is difficult and the results obtained very uncertain, so they are made of forms and dimensions that have been used in good practice and given good results. These fittings are bought in steel forgings. Boiler makers design the forged rings which lie close to the shell, of a section at least equal to the section of the plate that is cut out. The bearing surfaces of the manhole cover and that of the lip against which the cover bears, should be machined to make a good smooth joint. The joints are made tight by gaskets about $\frac{3}{4}$ of an inch wide.

Hand holes are constructed similarly to manholes, and often have a taper key in place of a bolt and nut, because the nut is exposed to fire and after it has been in place some time, is often difficult to remove with a wrench.

BRACKETS.

Boilers of the multitubular types are supported by brackets usually made of cast iron. Boilers up to 16 feet long have four brackets and those more than 16 feet long have six brackets. The brackets for this boiler should be about 10 inches long, measured with the length of the boiler, and about 15 inches wide. They are riveted to the boiler with nine or ten rivets $\frac{7}{8}$ to 1 inch in diameter. The rivets can be made large, as a large rivet makes a strong joint, and in this case the pitch is not governed by calking.

The load on the brackets can be estimated by calculating the weight of the boiler full of water and adding the weight of all the parts supported by the boiler. These parts include pipes, valves, gauges, brickwork covering, etc. This load should be divided as nearly equal as possible among the four brackets, so that the tendency of the boiler toward bending shall be small.

Brackets are set above the middle line of the boiler in order that the flanges may be protected by the brickwork setting. They are usually 3 or 4 inches above the middle.

CHIMNEYS.

At the present time, the knowledge of chimneys and chimney draft is slight. The theories given are worth but little as they are based upon data which is entirely insufficient. As to the design and proportions of chimneys, there are no systematic statements and rules that can be used.

Chimneys are usually designed from empirical formulas and from tables, compiled from proportions of chimneys that have furnished sufficient draft, etc.

The draft produced in a chimney is due to the difference in temperature, and consequently difference in pressure, between the gases inside the chimney, and the air outside. The gases in the chimney being lighter rise toward the top and air rushes in at the bottom to fill the space left by the hot gases. This air as it becomes heated grows lighter and rises, thus a continuous circulation is kept up. The temperature of the gases in the chimney is considered to be about 600° F. for chimney calculation, as practice shows this to give good draft under economical conditions.

After making several assumptions, based on experiments, the following formula has been deduced:

$$\text{H. P.} = 3.33\ (A - .6\sqrt{A})\ \sqrt{h}$$

in which H. P. = horse-power, A = area of the chimney, and h = the height above the grate.

The following table on page 46 has been calculated from this formula. This table is used to a considerable extent with satisfactory results.

The part of the table which is used for ordinary proportions is filled in. If proportions are taken from the table rather than from the formula, the results will give better proportions.

To find the area of the top of the chimney for a given coal consumption, the following empirical formula has been stated.

$$A = \frac{\text{H. P.} \times B \times 12}{\sqrt{h}}$$

in which A = area, H. P. = horse-power of boiler, B = number of pounds consumed per H. P. per hour and h = height of chimney in feet.

This area A is the area in square inches at the top.

Diameter, in Inches.	Height of Chimneys and Commercial Horse-power.											Side of square in inches.	Effective area, square feet.	Actual area, square feet.
	50 ft.	60 ft.	70 ft.	80 ft.	90 ft.	100 ft.	110 ft.	125 ft.	150 ft.	175 ft.	200 ft.			
18	23	25	27									16	0.97	1.77
21	35	38	41									19	1.47	2.41
24	49	54	58	62								22	2 08	3.14
27	65	72	78	83								24	2.78	3.98
30	84	92	100	107	113							27	3.58	4.91
33		115	125	133	141							30	4.48	5.94
36		141	152	161	173	182						32	5.47	7.07
39			183	196	208	219						35	6.57	8.30
42			216	231	245	258	271					38	7.76	9.62
48				311	330	348	365	389				43	10.44	12.57
54					427	449	472	503	551			48	13.51	15.90
60					536	565	593	632	692	748		54	16.98	19.64
66						694	728	776	849	918	981	59	20.83	23.76
72						835	876	934	1023	1105	1181	64	25.08	28.27
78							1038	1107	1212	1310	1400	70	29.73	33.18
84							1214	1294	1418	1531	1637	75	34.76	38.48
90								1496	1639	1770	1893	80	40.19	44.18
96									1876	2027	2167	86	46.01	50.27

Another method which is much more simple is to design the area of the chimney, as we have designed the total tube area; that is, about $\frac{1}{8}$ the grate area. This ratio for chimneys is sometimes about $\frac{1}{7}$ and decreases to $\frac{1}{9}$ and for very tall chimneys to $\frac{1}{10}$.

From the table we find the chimney to have an area at the top of about 3.98 square feet, assuming it to be 60 or 70 feet high. This area gives a diameter of 27 inches if circular, or 24 inches if square.

Let us calculate it from the formula $A = \frac{\text{H. P.} \times B \times 12}{\sqrt{h}}$

We must either assume or calculate B. As the calculation is very easy it would be better than any assumption. The total amount of coal burned per hour equals 12 × 24 or 288 pounds. The amount per H. P. per hour is 288 ÷ 75 or 3.84 pounds.

Then assuming the chimney to be 60 feet high,

$$A = \frac{75 \times 3.84 \times 12}{\sqrt{60}} = 446 \text{ square inches, or about 24}$$

inches in diameter if circular and 21 inches if square.

By the last method the area of the chimney will be 24 ÷ 8 or 3 square feet, or 432 square inches, giving practically the same result as with the formula.

As the table is reliable and gives us the larger area, we will use it and be on the safe side; also as the amount of coal burned per hour by the draft in a chimney can be found by multiplying the horse-power in the above table by 5, the chimney with an area of 3.98 square feet and 60 feet high will burn $72 \times 5 = 360$ pounds of coal per hour. The boiler in question burns only 288 pounds, so the chimney is sufficiently large..

Chimneys are usually of brick or of steel plates. If of steel they are always circular. When made of brick they are circular, square or hexagonal. With a given draft area, a circular chimney requires the least material, since a circumference has the least perimeter for a given area; it also presents less resistance to wind.

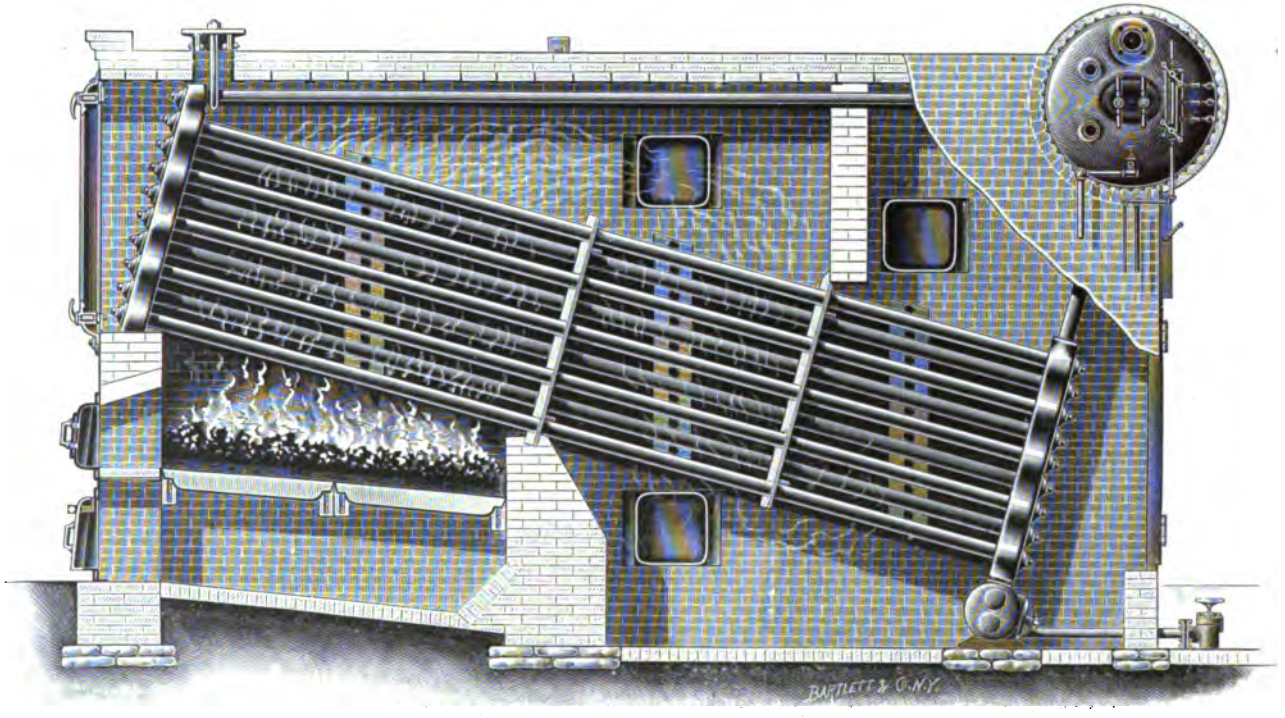
A steel chimney is made up of plates of steel riveted together. The shell is bolted through a foundation ring of cast iron to the stone foundation. It has a straight taper to the top, which is finished, for appearance with light plates. The shell is lined with fire-brick, with a thickness which varies from 12 to 18 inches at the bottom to about 2 to 4 inches at the top. This lining is used to prevent heat being lost from the shell and does not add to the strength of the chimney.

A brick chimney is built in two parts; *a* the outer shell, which resists wind pressure and *b*, the lining which is the flue. This flue is made separate from the external shell in order that it may expand, when the chimney is full of hot gases, without straining the outer shell.

The interior of both steel and brick chimneys are often cylindrical while the exterior tapers. The taper is about .3 inch to the foot. The brick at the base of the chimney is splayed out to make a large base.

As good natural earth should carry from 2000 to 4000 pounds per square foot, the base of the chimney should be large enough so that this pressure will not be exceeded.

The external shell is calculated for wind pressure and the weight of brick. This calculation for wind pressure involving higher mathematics will not be treated here. The lining is calculated for compression due to weight. The design, both of the chimney and its foundation, should be made by a competent engineer of experience, on account of disastrous results should a chimney fall.

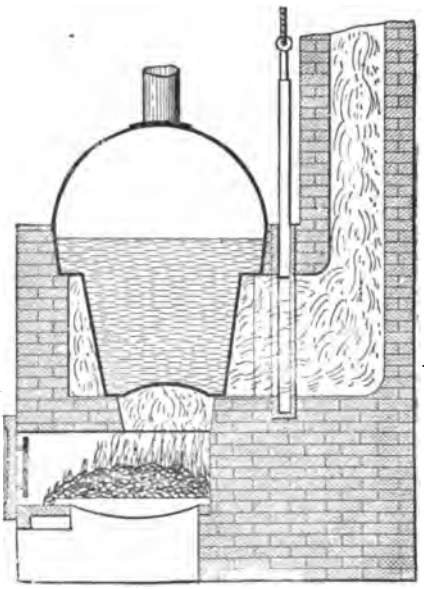


Side View of Cross Drum Type of Babcock and Wilcox Boiler.

SECTION 4.

TYPES OF BOILERS.

A Boiler is a vessel made to generate steam for power and heating purposes. All boilers have the following parts; the vessel or shell, which contains the water to be evaporated and sometimes the steam, a furnace in which the fuel is burned, a chimney to conduct away the smoke and to give draft to the fire, also various fittings, such as gauges, cocks, pipes, etc. The heating surface which transmits heat from the gases to the water, is usually part of the shell or the shell and the tubes within. The furnace is sometimes inside and sometimes outside the shell.



Haystack Boiler. Fig. 1.

Boilers are classified according to the work they are to do. The principal classes are Stationary, Locomotive and Marine.

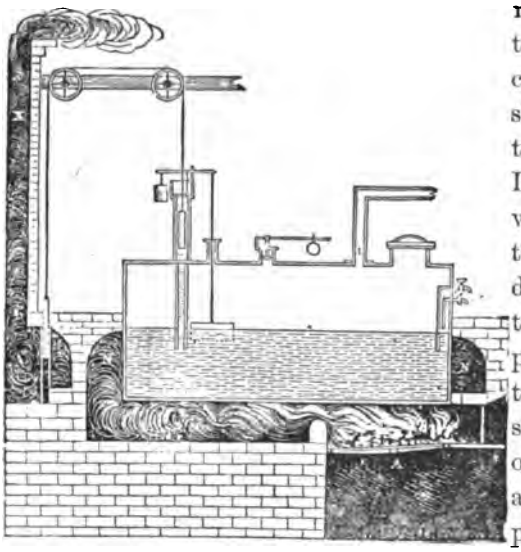
We will not enter into the early history of the steam boiler very fully here, but in order to understand the growth and developments we will consider a few early forms.

Newcomen's Spherical Boiler. One of the earliest forms was a spherical boiler designed by Newcomen about 1710. It was made of cast iron and a brick-setting was so arranged that the hot gases from the furnace, which was directly under the sphere passed through a flue, which surrounded the boiler just below the water line, and then up the chimney opposite the furnace.

The Haystack Boiler. Newcomen also designed a vertical cylindrical boiler, called from its shape the "Haystack" or "Balloon" boiler. This boiler was made of wrought iron with a hem-

ispherical top and arched bottom as shown in Fig. 1. In 1769, Smeaton improved this boiler by placing the fire inside the shell and conducting the hot gases to the chimney by internal flues. This arrangement increased the heating surface.

Wagon Boiler. On account of its greater heating surface, a horizontal boiler designed by Watt, replaced the haystack boiler. From its resemblance to a wagon top it was called the "Wagon" boiler, Fig. 2. The top was hemispherical, and the bottom curved inward. The products of combustion passed from the grate underneath the boiler to the rear, then through the left-hand flue to the front, and after passing by the front and returning through the right-hand flue passed to the chimney. This is called the wheel draft since the gases pass entirely around the boiler. In the large sizes, a flue was placed in the boiler through which the products of combustion returned to the front after passing under the boiler to the rear, as in the small sizes. The gases, on issuing from the flue at the front, divide and pass to the chimney at the rear of the boiler by the flues in the brickwork on either side. This form of draft is called the split draft. Watt used a column of water in the feed pipe, which was vertical, as a pressure gauge. The rise and fall of this column also controlled the damper. The feed water was regulated by a float. Because of its shape and the tendency to change its shape even with careful staying, this form is not suitable for pressures now used.



Wagon Boiler. Fig. 2.

These boilers were satisfactory because low pressures were used, but when a higher pressure than 15 pounds per square inch

above the atmospheric pressure was demanded, a stronger form was necessary.

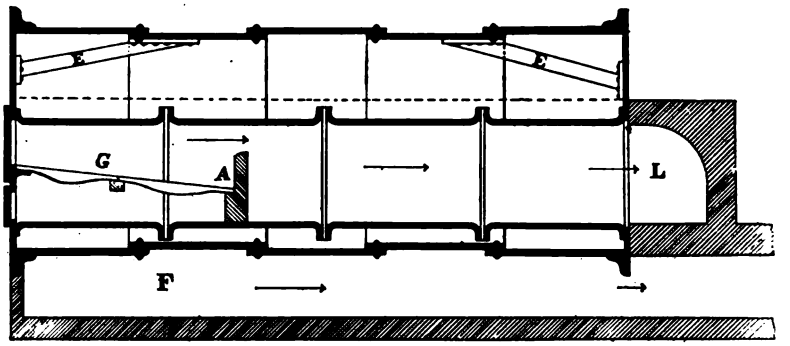
MODERN BOILERS.

About the beginning of the present century the modern type of boiler was introduced. One of the earliest forms was the plain cylindrical boiler of wrought iron, with flat or hemispherical ends. This latter form was called the "Egg end" boiler. The flat end is made of cast iron, and the hemispherical end of wrought iron. The egg end boilers are simple to make and require no staying because their form is, withthe exception of the sphere, the strongest to resist rupture. The heating surface is small with a given diameter, unless the boiler is made very long, and all the sediment collects in the bottom where the heat is greatest, which causes the plates to burn. Sediment and scale are poor conductors of heat. These disadvantages are partly made up for by its simplicity, ease of repairing, inspecting and cleaning, and its strength and durability. They were extensively used in places where fuel was abundant, as in mining districts, in blast furnace and iron works. These boilers are 30 to 42 inches in diameter and from 20 to 50 feet long, sometimes even longer.

In order to increase the heating surface of the plain cylindrical boiler, without increasing its dimensions, an internal flue through which the hot gases passed to the chimney, was introduced. At first this type had only one flue of large diameter extending from end to end. This form was called the cylindrical flue boiler. Two improvements were made on this boiler, one of which developed the internal flue boilers, and the other the multitubular boilers.

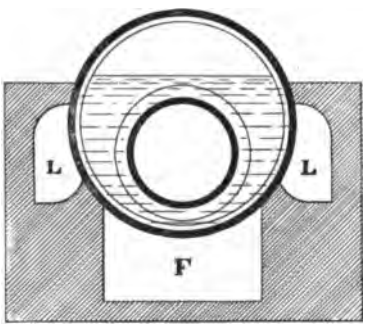
The Cornish Type. When it was discovered that about 25 per cent. of the total heat of combustion was lost by radiation from the furnace, a Cornish engineer, named Trevithick, conceived the idea of placing the fire inside the internal flue. He introduced this boiler which is now known as the Cornish boiler, Fig. 3. The products of combustion pass from the fire grate G through the flue to the back end of the boiler, where they divide and return to the front end by the lateral flues L in the brickwork. At the front the products of combustion pass downward and uniting in

flue F, go to the chimney in contact with the bottom of the boiler. This arrangement of flues reduces the temperature of the gases before they come in contact with the bottom of the boiler where the sediment collects.



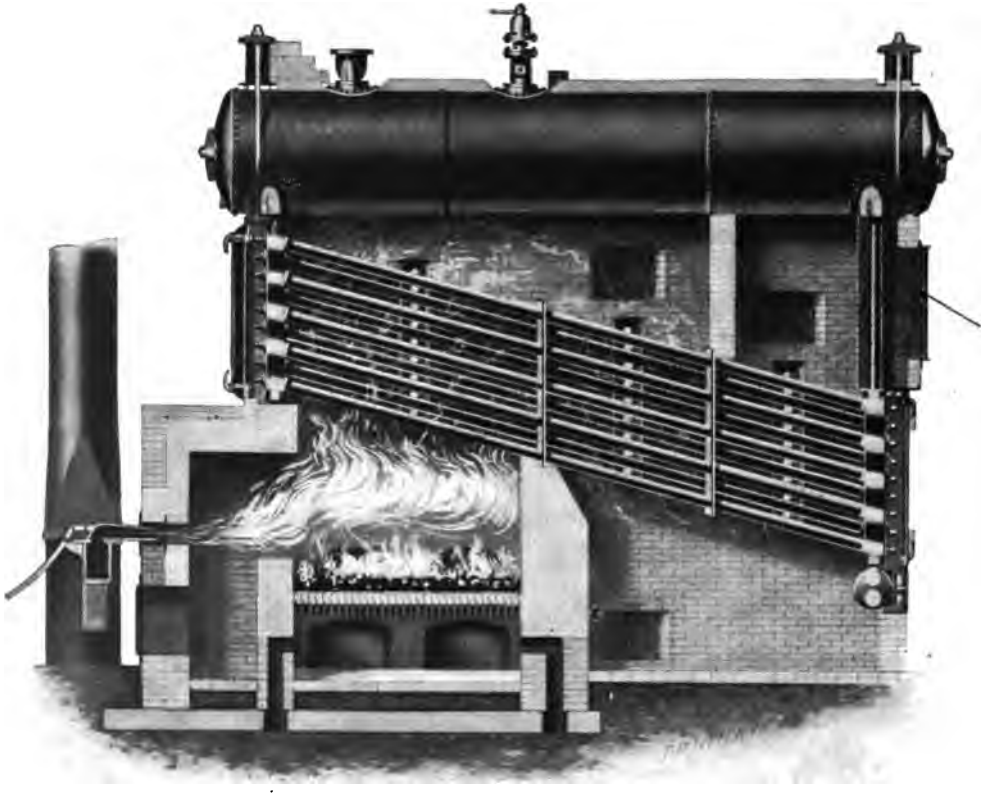
Cornish Boiler. Longitudinal Section. Fig. 3.

The internal flue is the hottest portion of the boiler, and consequently has a greater linear expansion than the shell. This causes a tendency to bulge the ends. When the boiler is cold the flue returns to its normal length and has a tendency to work loose from the ends. To remedy this the flues are made up of a number of short rings, joined together by being flanged and riveted to a plain ring as shown in Fig. 3; or by having the plain rings riveted to a ring shown in section in Fig. 4. This curved ring serves as a spring and allows of expansion and contraction and increases the heating surface. It also stiffens the flue against external pressure. Another method to obtain the same result, is the use of corrugated flues, Fig 5. We will consider these flues later in marine boilers.



Cornish Boiler. Transverse Section. Fig. 3a.

The grate bars rest at one end on the dead plate and at the other on the bridge A; if they are made in two lengths, they are supported by a cross bearer at the center. The furnace bridge A



Section of Babcock and Wilcox Boilers Fitted with Kennedy Burners for Burning Blast Furnace Gas.

is built of fire brick and the external flues are usually lined with fire brick. The head is stayed to the shell by gusset stays E.

Lancashire Boiler. Large flues are liable to collapse under external pressure. On this account and because of the short distance which a fireman can throw coal accurately, the Cornish boiler can be used for small powers only. When large power is required a modification of the Cornish boiler is used. This boiler is called the Lancashire boiler and is shown in Fig. 6. It is like the Cornish, except that it has two flues and two furnaces. This allows smaller, and therefore stronger flues for a given power, that is the boiler can be made larger without danger of collapsing the flues.

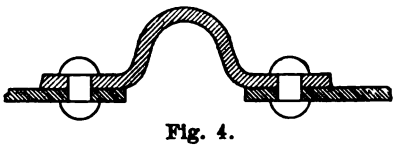


Fig. 4.

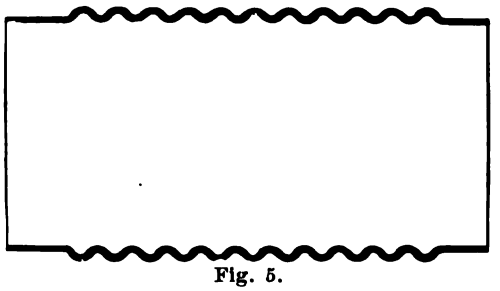
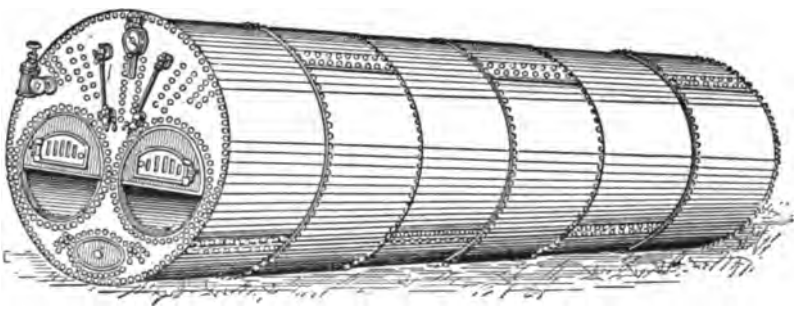


Fig. 5.

The flues are sometimes continued separately to the end. If they merge into one large flue, which forms the combustion chamber, it is called a "Breeches Flued" or duplex furnace boiler. These



Lancashire Boiler. Fig. 6.

furnaces are fired alternately and the unburned gases from the fresh coal meet the hotter gases from the incandescent coal of the other furnace, and are burned. This arrangement prevents the escape of the unburned hydrocarbons, and also

sudden changes of temperature (thus giving the boiler high efficiency). The disadvantage of the Lancashire boiler, is the difficulty of finding room for two flues without increasing the diameter of the boiler. Small flues are stronger, but the low furnace is unfavorable to complete combustion. The combustion chamber overcomes this objection. The construction at the junction, or neck, of the two tubes is necessarily weak and is the source of many disasters.

Galloway Boiler. In order to increase the heating surface, strengthen the internal flue and improve the circulation of flue boilers, small conical tubes which act as pillars, are added. These water tubes are called Galloway tubes, and the boiler, the Galloway boiler. Fig. 7 shows the general method of inserting the tubes in the flue. They are made conical (or of less diameter at one end than at the other) to facilitate repairs. These tubes are often welded to the flue to prevent leaking; but if one must be replaced, the tube and welded part must be cut out leaving a large hole in the flue. On this account they are often riveted instead of welded. These tubes in addition to other advantages, break up the currents of hot gases and cause them to give up heat by obstructing the flow. They are often placed in Cornish and Lancashire boilers. The Breeches boiler especially is much improved by the Galloway tubes.

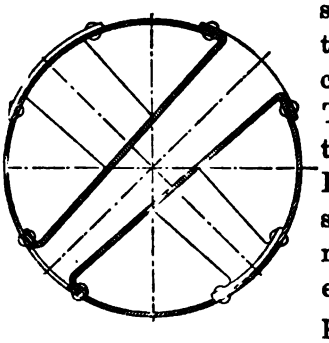


Fig. 7.

Multitubular Boilers. When it was found that the internal flue was such an advantage (that is, it increased the heating surface), boiler makers soon added other tubes and as the number increased the size diminished until they became of the size used at present. This is briefly the development of the externally fired multitubular boiler. This type of boilers is shown in Figs. 8 and 9. The boiler without the brick setting is shown in Fig. 8. It consists of a wrought iron or steel cylindrical shell, and fire tubes terminating in the two flat ends. These tubes are three or four inches in diameter and are expanded in the tube sheets or ends as

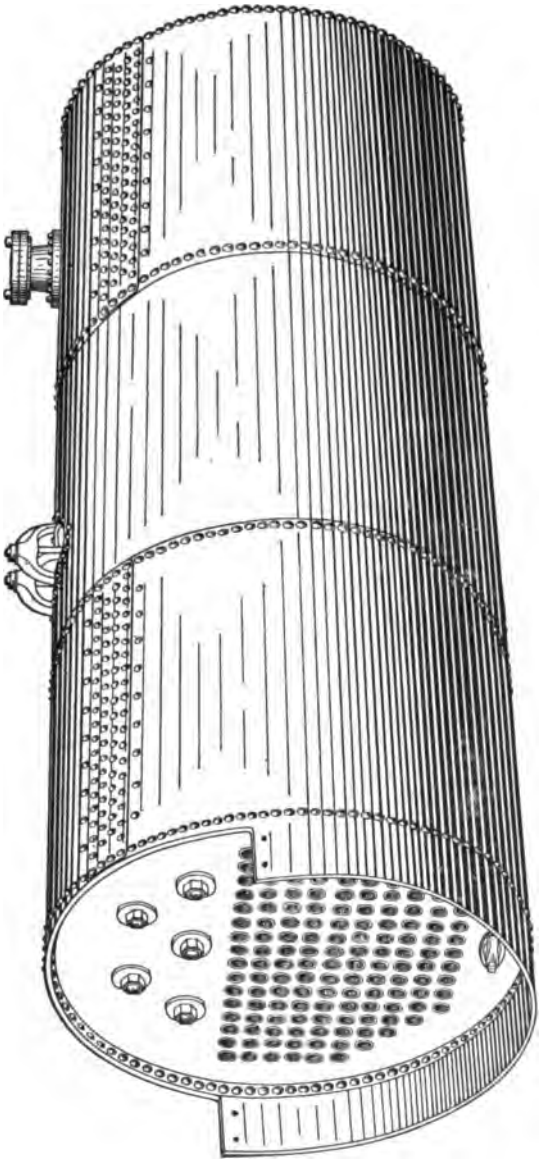
explained in "Construction of Boilers." About one-third the volume of the boiler is occupied by the steam; this space may be made a little less if a steam dome is used. The other two-thirds is the water space in which are the tubes; the water line being a few inches above the top row to prevent burning of the same. The end plate below the water line is stayed by the tubes; above the line the plate is stayed by through stay rods, or diagonal stays.

At the front of the boiler the shell extends beyond the tube plate and is cut away to allow the products of combustion to enter the uptake. The boilers shown in the figures are supported by brackets riveted to the shell. The front bracket is often fixed in the side wall but the rear bracket should be so arranged that it may move in a direction with the length of the boiler, so that the shell will not be strained by expansion of the plates. The rear bracket is often placed on rollers which move on an iron plate. A small space is left between the rear tube sheet and the brick wall, to allow for expansion.

The boiler shown in Fig. 8 has two steam nozzles. If steam is taken from a dome, the steam nozzle is at or near the top of the dome.

The feed pipe may enter either at the front or rear. It usually terminates in a perforated pipe below the water line. The blow off pipe is at the rear of the boiler and leads from the back tube sheet. A boiler is usually set with a slight inclination toward the rear so that mud and detached scale will collect near the blow off pipe. This pipe is also used to empty the boiler. The blow off pipe has a valve which may be opened when steam is up, to blow out sediment.

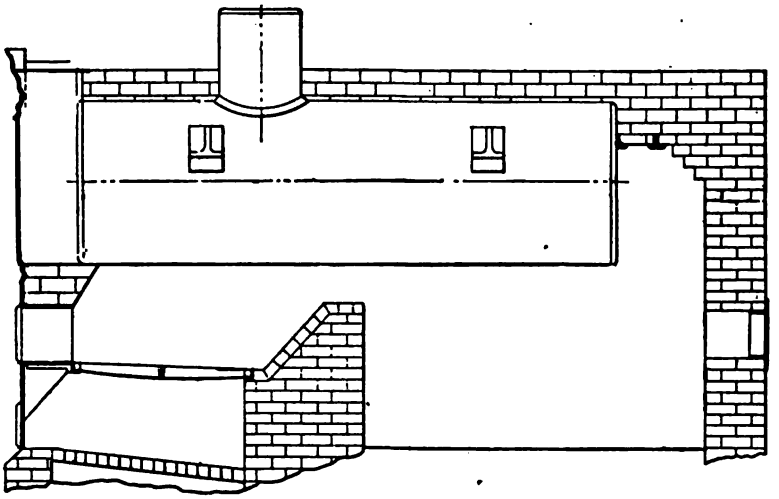
In order that the boiler may be washed out and entered for cleaning or repairs, it is provided with manholes and handholes. Fig. 8 shows a manhole at the top near the middle, and a handhole near the bottom of the tube sheet. Handholes may be put in wherever desired, but manholes can only be located where the arrangement of stays and tubes will permit the entrance of a man. In this type of boiler, the heating surface is composed of about half the shell, the inside of the tubes and about two-thirds of the rear tube sheet, since these surfaces come in contact with



Multitubular Boiler. Fig. 8.

the products of combustion. All the heating surface is below the water line.

Fig. 9 shows a complete multitubular boiler in its brick setting. The brick setting is made of brick laid in cement or mortar. The bridge and portions of the walls at the front end and all parts exposed to the fire are lined with fire brick. The bridge is



Multitubular Boiler Setting. Fig. 9.

built at the rear of the grate bars and thus supports the grate bars and directs the flames upward. The furnace is formed by the side walls, the bridge and the lower part of the boiler front, which is usually of cast iron. Its lower part is lined with fire brick. Doors are placed in the rear and side walls to facilitate cleaning. The front is made of cast iron with doors leading to the furnace, ash pit and smoke box. The space below the grate is called the ash pit, and through its doors ashes are removed and the air furnished for combustion. Both the fire doors and the ash pit doors have draft plates to regulate the supply of air. The doors to the smoke box give access to the tubes.

A stationary boiler which combines the internal flues of the Lancashire and the fire tubes of the multitubular is sometimes used. It is short, of large diameter, with corrugated flues like the Lancashire, but the space above the flues contains fire tubes which are parallel to the flues.

The heated gases pass through the flues to the combustion chamber, which is at the rear and lined with fire brick; then return to the front end by the fire tubes. The uptake is at the front as in the multitubular boiler. This type of boiler has a large heating surface in proportion to its grate area.

The fittings and stays are similar to those of the Lancashire and Multitubular boiler.

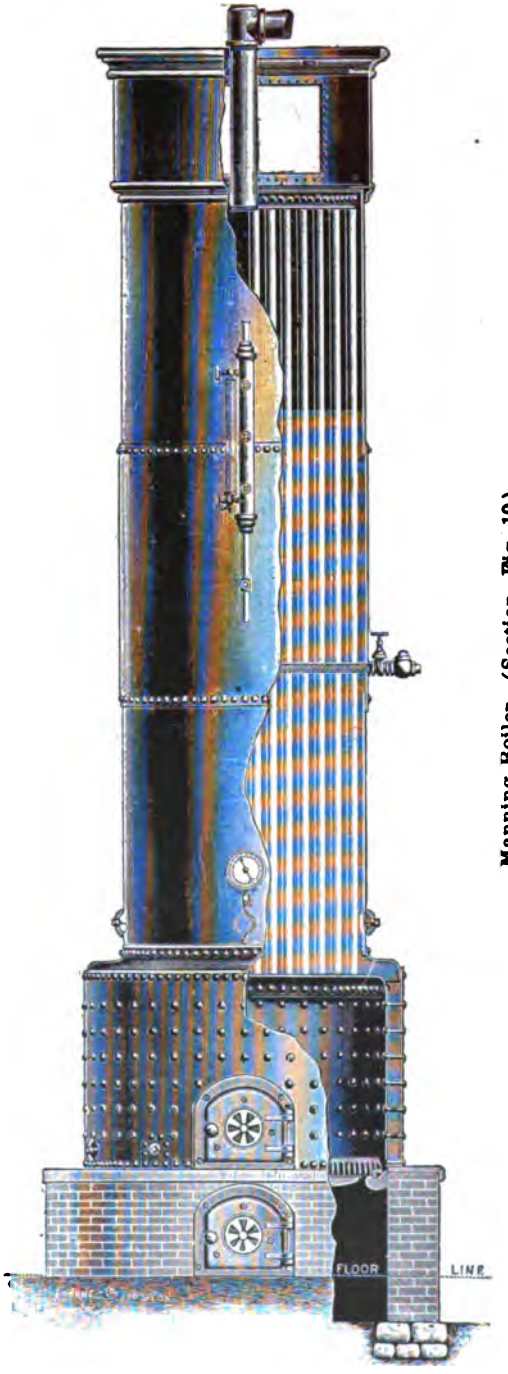
The boiler is enclosed in brick work which is not the support, but simply a covering to prevent loss by radiation. The boiler itself is supported on wrought iron beams.

Vertical Boilers. Vertical boilers are used when floor space is valuable and there is sufficient height. In general they are not as economical as horizontal boilers unless carefully designed. They are used in small sizes, for hoisting, pile-driving and similar work; and in large sizes where it is necessary to have a powerful battery of boilers in a small space.

They have a vertical shell of cylindrical form with a fire-box at the lower end, and fire tubes connecting the furnace with the top of the boiler. Masonry foundations and brick ash pit are used with large sizes, but for small portable boilers the cast iron foundation serves as the ash pit.

Fig. 10 illustrates a vertical Manning boiler. In order to get a large heating surface it is made from 20 to 30 feet high, and the fire-box is surrounded by water. The tubes are arranged in concentric circles with a space for circulation in the middle. This space in the upper tube sheet permits the attachment of the steam outlet.

At the bottom, the shell is of greater diameter than at the top to provide a large furnace and grate area. It is stayed to the outside shell by screw stay bolts thus forming small squares, as shown in the engraving. These bolts are screwed through both shells and riveted over on both ends, and have a small hole drilled from the outside nearly to the other end. If a stay should break the escaping steam or water would warn the fireman. The external fire-box shell is joined to the shell by a double flanged ring, shown in Fig. 10 or by the cone shaped section, as in Fig. 11. The top of the internal shell is riveted to the lower tube sheet which is flanged. The bottom of the internal fire-box is connected to the



Manning Boiler. (Section. Fig. 10.)

Fig. 11.

external fire-box by a welded iron ring called the foundation ring. The water space between the fire-box plates, called the water leg, should be made large.

The only provision for cleaning this boiler is by means of handholes in the shell near the lower crown sheet, and in the external fire-box near the foundation ring. For this reason only good feed water should be used.

The feed water pipe enters the shell at the side near the mid-

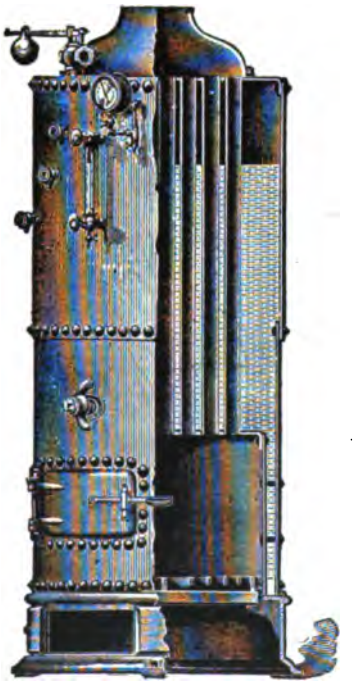


Fig. 12

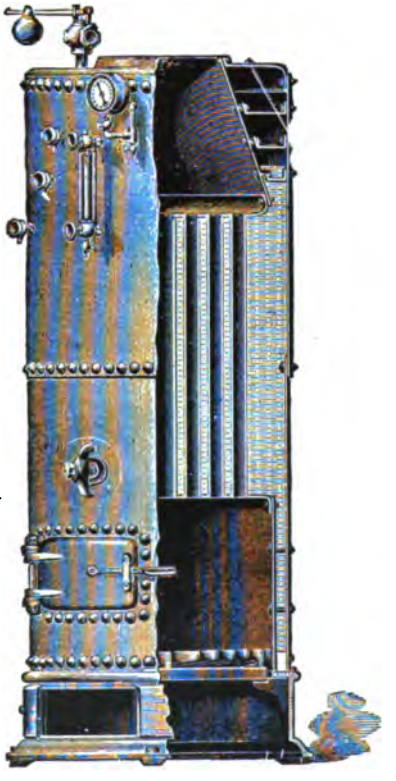


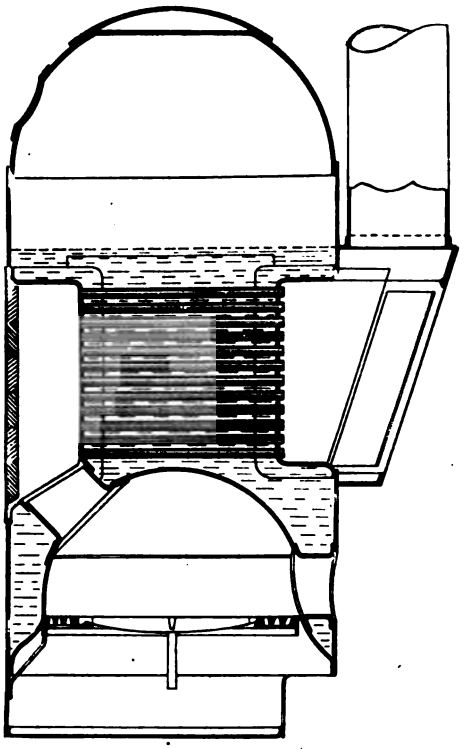
Fig. 13.

dle of the water space, and extends across the boiler; it is perforated to distribute the water.

The heating surface consists of the inside of the tubes up to the water level, the tube sheet, and the sides of the fire-box. The inside of the tubes above the water line is the super-heating surface. Heat is transmitted from the gases to the steam; raising its

temperature without raising its pressure. Steam thus heated is called super-heated.

Small vertical boilers, used for light work are constructed as shown in Figs. 12 and 13. They usually have a short smoke pipe into which the exhaust steam may be turned. The boiler shown in Fig. 12 is a common form; the tubes extend from the fire-box to the upper head, passing through the steam space. These tubes, unless of small diameter, are liable to collapse if high pressures are used, and are also liable to injury from overheating. The upper ends of the tubes of the form shown in Fig. 13 are below the water level, to avoid the difficulties stated above. The upper submerged tube sheet is flanged and riveted to the cone which forms the smoke chamber. The space between the shell and the cone is apt to be small, and not sufficiently large for the steam formed on the tubes, to rise to the steam space. The cone which is subjected to external steam pressure is likely to be weak.



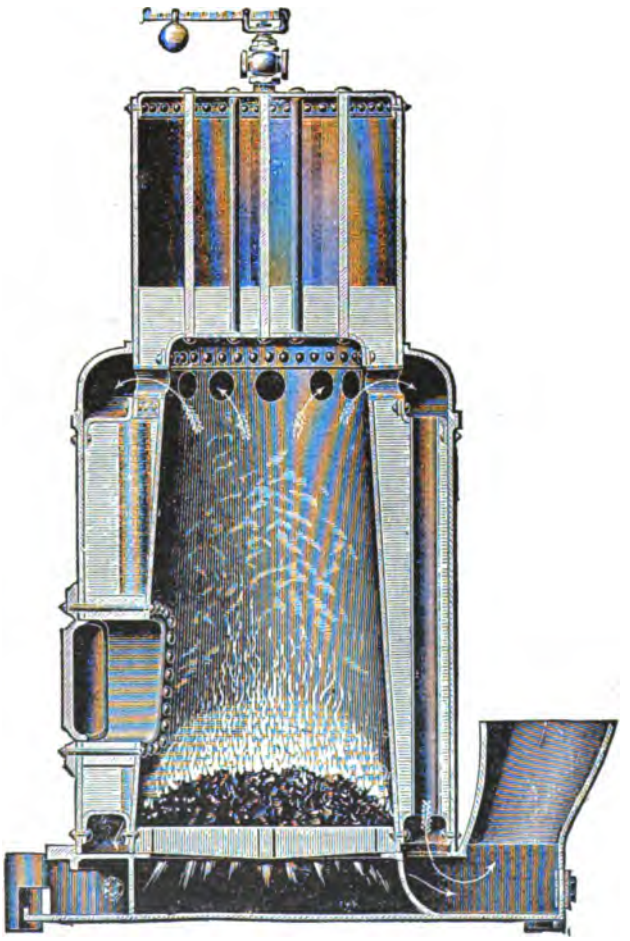
Cochrane's Vertical Boiler. Fig. 14.

Cochrane's Vertical Boiler is similar in the arrangement of heating surface to a marine boiler. A sectional view is shown in Fig. 14. The hot gases go up one side to the combustion chamber, then through the horizontal tubes to the uptake. The shape of the boiler and arrangement of tubes gives a large heating surface and steam space. The crown of the boiler and of the fire-box being hemispherical, require no staying.

Babcock and Wilcox Marine Water Tube Boiler.

The flat surfaces, or the tube plates, are held together by the tubes.

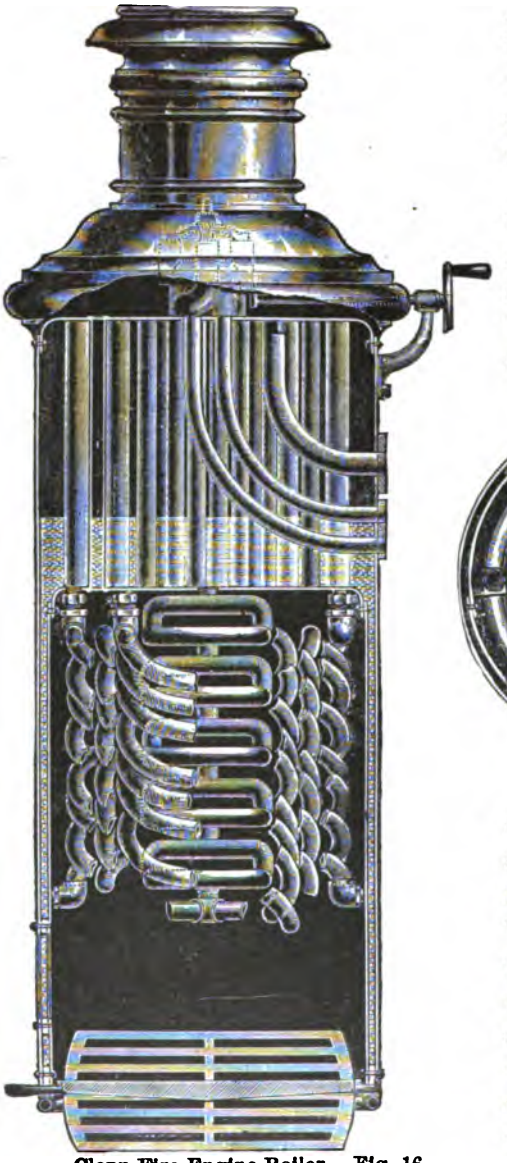
The Shapley Boiler, shown in Fig. 15, is a return flue vertical boiler. It is made in two parts, the upper being a reservoir for water and steam, while the lower one, which is larger, contains the fire-box. The top or crown sheet of the fire-box is stayed to the top by through stay rods. Near the top of the fire-



Shapley Boiler. Fig. 15.

box a number of short tubes connect with an annular space which is connected with the base flue by vertical tubes passing through the water space. This boiler is very durable, because the crown sheet and tubes are away from the fire. All the surfaces in con-

tact with the fire are covered with water, which makes a large heating surface and prevents burning of the plates. The base is so constructed that it can be partially filled with water, which quenches any sparks that may be carried over.



Clapp Fire-Engine Boiler. Fig. 16.

The Fire-Engine Boiler. Steam fire-engine boilers are vertical and self-contained. They are light, compact and able to make steam

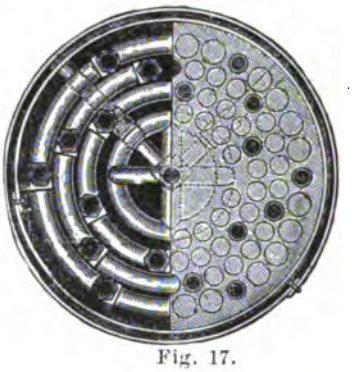


Fig. 17.

quickly and in considerable quantities when forced draft is used. The forced draft is obtained by turning the exhaust steam into the smoke pipe. These boilers are often a combination of the tubular and water-tube types. Fig. 16 shows a section of the Clapp fire-engine boiler. It has a cylindrical shell with a deep internal fire-box at the bottom, which is surrounded with water.

The water tubes are arranged in three concentric sets of helical coils, with the central space occupied by a series of castings of the form shown in the engraving. The top of each casting is shaped like an inverted U and the lower part is circular with a cross tube. The products of combustion from the furnace rise among these tubes until they reach the fire tubes, through which they pass to the smoke pipe. Half of Fig. 17 shows the helical tubes from above and the other half the arrangement of the fire tubes.

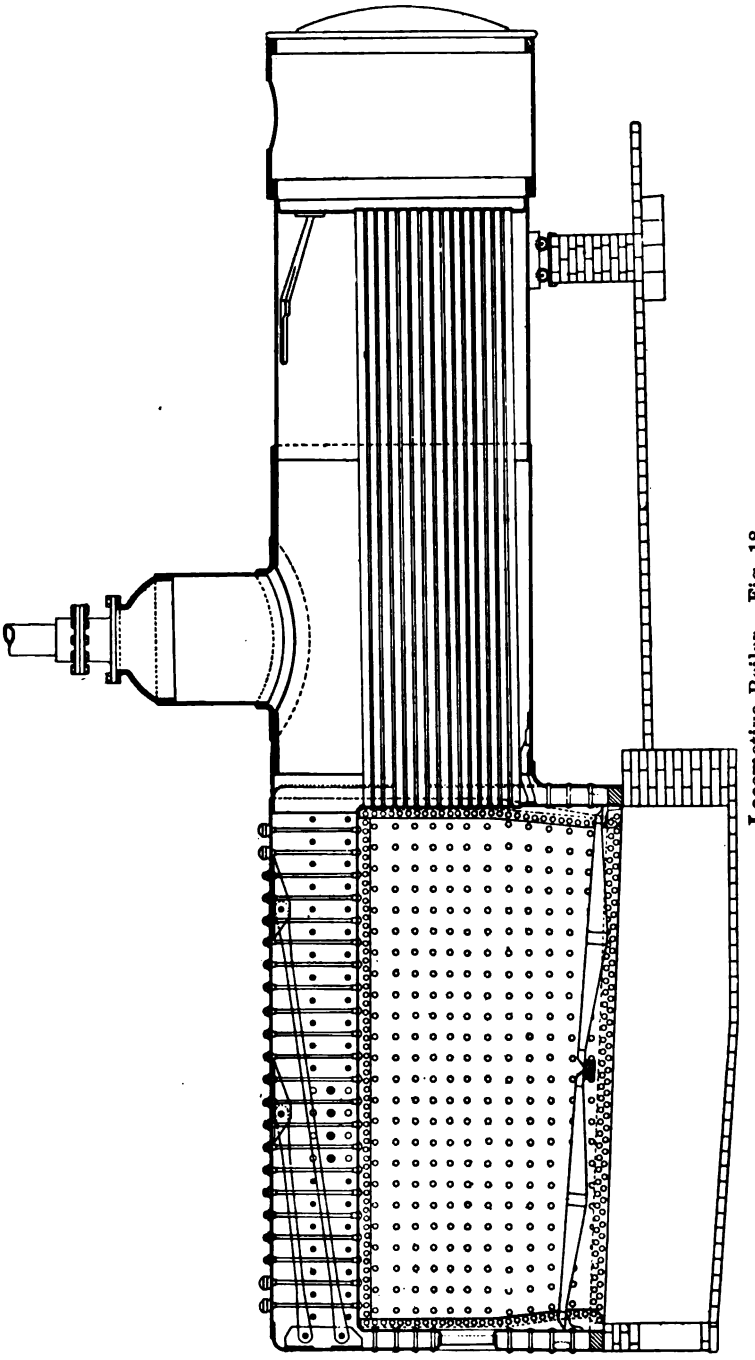
THE LOCOMOTIVE TYPE.

The locomotive or fire-box boiler Fig. 18, belongs to the non-stationary or portable class. It is self-contained, that is, it requires no brick-work for flues or setting. It is used in railway service, as a stationary boiler, and for agricultural purposes.

For stationary work it consists of a cylindrical shell and a rectangular fire-box which are joined together by riveting to a flanged ring similar to that used in a vertical boiler or by the use of a cone-shaped section. The fire-box has a rectangular cross section with vertical sides and usually a semi-cylindrical top. Like the vertical boiler there is an inner and outer fire-box; the inner having the same shape as the outer, already described, except that the top is flat. The external fire-box is connected to the inner by short screw stays as described in "Construction of Boilers." The space between is called the *water leg*. The flat top is strengthened by parallel girders which are sometimes attached to the shell by sling stays. A large number of small fire tubes extend from the front sheet of the furnace to the front head or tube plate. These tubes act as stays for the lower portions of the plates; the top portions being stayed to the shell by diagonal stays.

The shell is prolonged beyond the front tube plate to form a smoke box. The products of combustion from the fire pass through the tubes to the smoke box, from which they pass to the smoke stack or uptake.

This boiler is supported at the fire-box end on a brick foundation which forms the ash-pit, or by brackets; at the front end by a pedestal bracket or cradle with rollers. This allows for the slight expansion of the plates. Several handholes are placed in



Locomotive Boiler. Fig. 18.

the water leg to facilitate cleaning and it is usual to put a man-hole at the top.

Steam is taken from a steam dome on which is the safety valve. The feed water may be introduced at any convenient place, but it is better for it to enter at some distance from the steam dome. The pressure gauge, cocks and water glass are placed on the rear plate of the external fire-box.

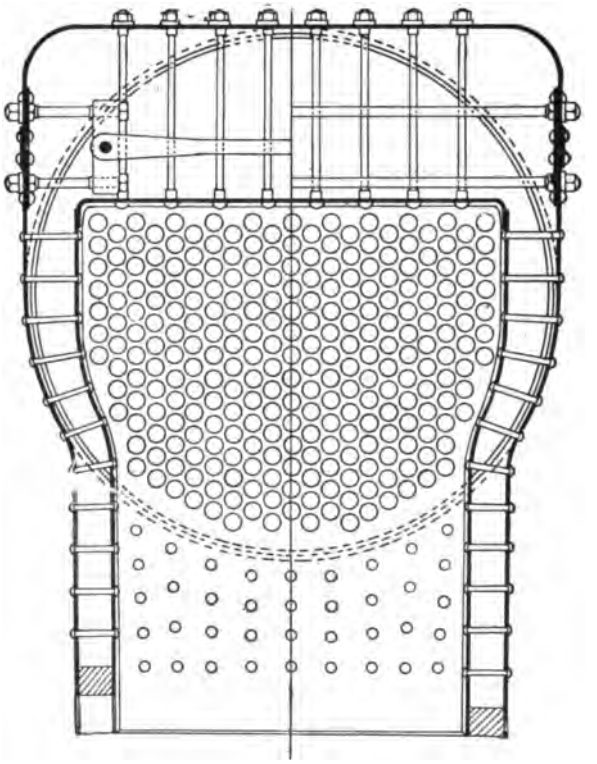


Fig. 19.

For locomotives, the tubes are made smaller, about 2 inches in diameter and put near together to get a large heating surface, and to obtain rapid combustion the exhaust steam is turned into the smoke bo. . The blast of steam carries off the hot gases, causing a new supply of air to enter the ash-pit. The rate of combustion per hour often exceeds 100 pounds of coal per square foot of grate surface. The boiler is fastened rigidly at the

smoke box; the frames which support the fire-box end are attached to it by expansion pads to provide for the longitudinal expansion.

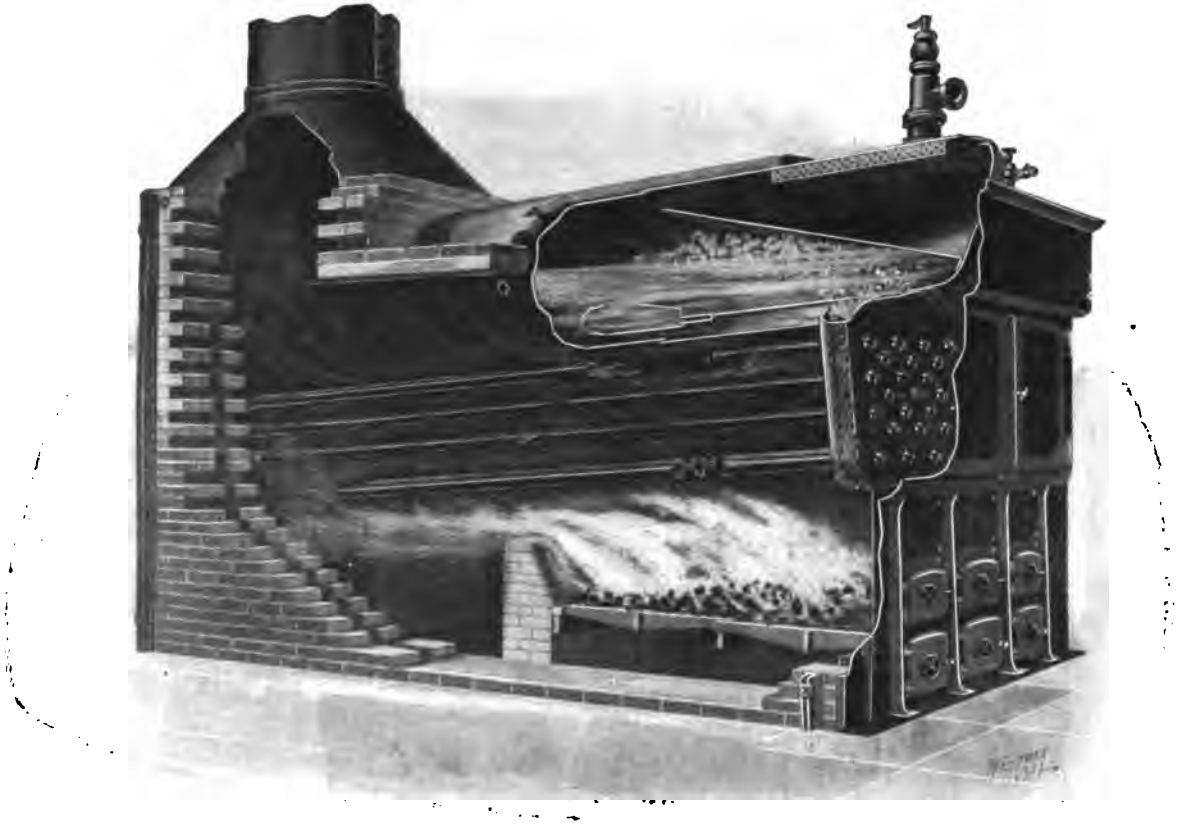
The boilers already considered are composed of a shell, containing steam and water, and internal flues through which the products of combustion pass. The chief defects of these boilers are, liability to disastrous effects from explosion, weight, size, difficulty of transportation, and slowness of raising steam. There is a type of boilers which we will now consider that has been designed to overcome these defects. They are called water-tube or tubulous boilers to distinguish them from the tubular type. The main difference is that the tubes of the water tube type contain water, with the hot gases on the outside, while in the tubular boilers the hot gases pass through the tubes which are surrounded with water.

WATER TUBE BOILERS.

The water-tube boiler with its various forms and methods of construction has struggled for success for several years and at present holds a prominent place among boilers that are most extensively used. The principal reason why they are not more generally adopted is that considerable engineering skill is required to make them successful. Water-tube boilers, if well designed and well constructed, are light, powerful, compact and well adapted for forced draft.

The many forms differ in detail and method of construction, but all have the following parts and principles. The steam is separated from the water and stored in a drum or reservoir, which is comparatively large in order to secure a storage capacity for hot water and a large surface for the disengagement of the steam. A large portion of the heating surface is composed of tubes of small size, containing water and steam. The tubes are connected by pipes to the steam and water drum and also to a mud drum at the bottom. The hot gases from the fire are conducted among the tubes and after losing a portion of their heat they come in contact with the steam drum.

The following are the general advantages of water-tube boilers.



Heine Boiler, Showing Travel of Gases and Method of Setting.

The heating surface next the fire facilitates the transmission of the heat and prevents overheating. The draft area, which is limited in fire-tube boilers, is the whole chamber in which the tubes are enclosed. This allows plenty of time for the heat to be absorbed from the gases before they pass into the chimney. Riveted joints are removed from the fire; a desirable feature, since the double thickness of metal when exposed to the direct action of the flame causes serious difficulty, and owing to their weakness, all strains of unequal expansion are concentrated there.

Since the steam enters one end of the steam drum and leaves it at the other end, the water and steam are completely separated.

As all the water tends to circulate rapidly in one direction, there is little chance for sediment to be deposited on the heating surfaces, and the parts are kept at nearly the same temperatures.

Steam may be raised rapidly, as the water is divided into small streams which pass through the hottest part of the furnace.

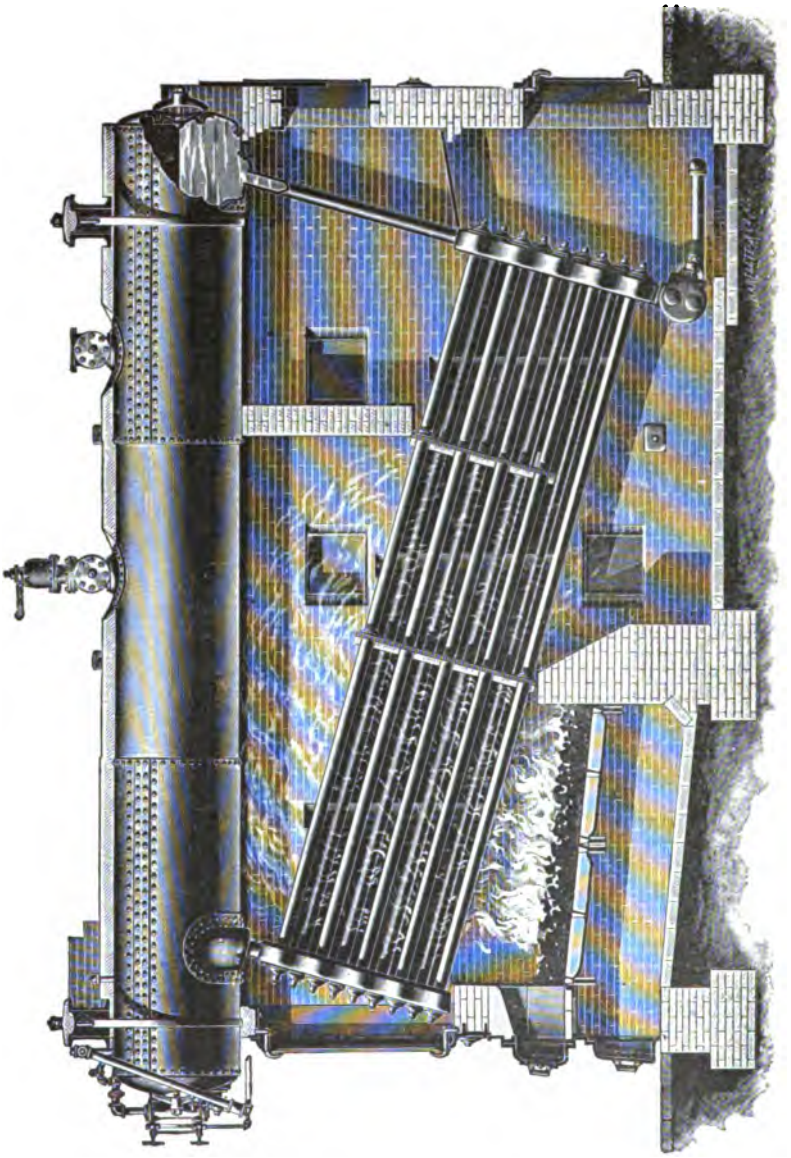
The loss of heat from dust and dirt is smaller than in a *fire-tube* boiler, because little dirt will remain on the convex surfaces of *water tubes*, which then become self-cleaning. The inside of *fire tubes* are quickly covered with dirt, and have to be cleaned frequently.

This type of boiler being made in sections is readily taken apart and transported from place to place.

One of the greatest advantages of these boilers is their safety. On account of the good circulation, no part exposed to the fire is uncovered while there is any water in the boiler. There cannot be overheating until the boiler is empty, and even then there would be no serious explosion. The water is divided into small masses, which prevent serious destructive results in case of rupture, since the thermal storage is small.

Babcock and Wilcox Boiler is one of the best of the water-tube type. Fig. 20 shows a sectional view and the general arrangement of drum, tubes, fire, brickwork, etc. A partial section of the tubes and steam and water drum is shown in Fig. 21.

This boiler is composed of lap-welded wrought-iron tubes, placed in an inclined position, and a horizontal steam and water drum above them. The tubes are connected with the drum by nearly vertical tubes at either end. This steam and water drum is



Babcock & Wilcox Boiler. Fig 20

made of wrought iron or steel plates, extra thick and double riveted. The mud drum is made of cast iron, as that is the best material to resist corrosion. The tubes in the vertical rows are "staggered," that is, each tube in a horizontal row comes over the space in the row below, so that the current of hot gases will be broken up.

The boiler is suspended from wrought iron girders, which rest on iron columns, and is entirely independent of the brickwork. This permits repairing and removing of the brickwork without disturbing the boiler.

As will be seen from the engraving the fire is under the front and higher end of the inclined tubes. The products of combustion are guided by division plates and bridges so that after rising from the fire grate, they pass between the tubes, to the combustion chamber, which is under the drum, then downward among the tubes and, after rising between the tubes for a second time, they pass up the chimney.

This method of conducting the gases and staggering the tubes causes almost all the heat to be absorbed from the gases.

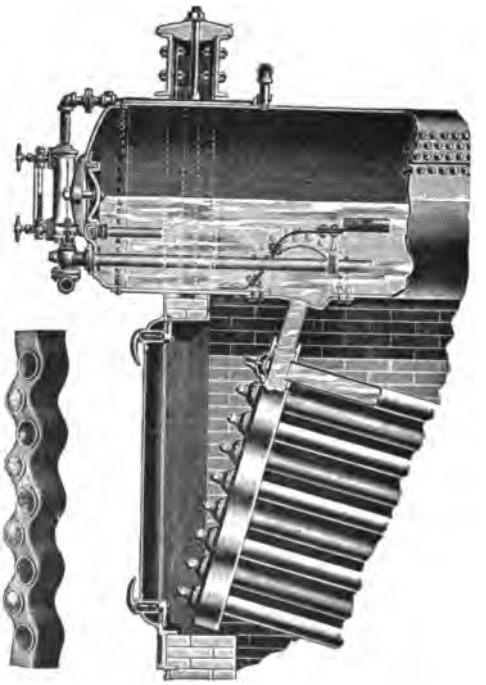
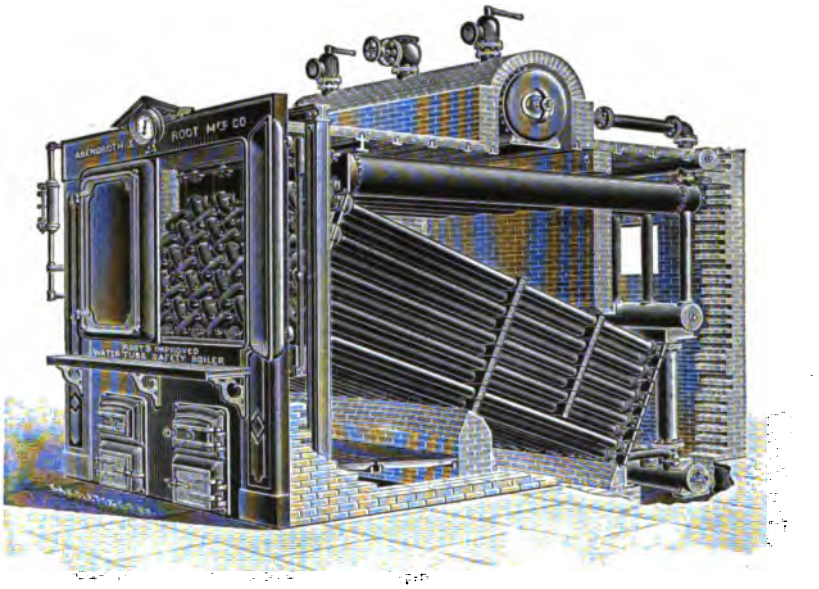


Fig. 21.

As the water in the tubes becomes heated, it rises toward the higher end where it becomes partly converted into steam, and a column of water and steam rises to the steam and water drum, in which the water and steam become separated. The water flows through the drum and down the tubes at the rear, thus effecting a good circulation, which is quite rapid, as the tubes are all large.

These boilers have gauges, hand-holes, safety valves, etc., the same as fire-tube boilers. They are well adapted for use over puddling and blast furnaces, where the waste gases are used as fuel, and have also been used successfully in rolling mills, where the work is heavy and irregular.

The Root Water Tube Boiler. Fig. 22, is similar to the Babcock and Wilcox. It is composed of four-inch wrought-iron, lap



Root Water Tube Boiler. Fig. 22.

welded tubes, expanded in pairs, as shown in A, Fig. 23, into cast iron headers at both ends. These pairs are placed one on the other, forming a vertical section, as in Fig. B. Each tube is connected to one in the pair above it by a flexible bend, by means of which a perfect and uninterrupted circulation of water from the bottom to the top of the section is obtained. To form the boiler, several of these vertical sections are placed side by side, a metallic packing-ring insuring a tight joint between the bend and the header, as shown in detail in Figs. D, E and F. The tubes in the vertical section are staggered, while the pairs or packages are so arranged that they are not rigid, for it is well known that the

lower tubes being nearer the fire will expand more than those above.

Each section has its own overhead drum, to receive its discharge of water and steam. At the rear of the boiler and at the end of the overhead drum, there is a vertical pipe leading to a drum beneath, which lies at right angles to the tubes and serves as a common reservoir for all the sections. Into this drum the feed water enters and meets the hot water from above. The mixing of the water causes a sufficiently high temperature to prevent any trouble from unequal expansion. This cross drum is also connected by vertical pipes to another drum below and parallel to it, which is the mud drum. From the cross feed drum, the mixture of circulating and feed water descends through the large vertical pipes to the mud drum beneath. After leaving the solid impurities, the clear water rises to the top of the mud drum and flows into the extreme lower end of each vertical section of water tubes. In these tubes, which are in contact with the hot gases from the fire, the water is heated hotter and hotter as it rises to the upper ends. The water, filled with bubbles of steam, passes from the tubes to the overhead drums, where the steam and water are separated; the steam going to a common steam drum at right angles to these drums and the water descending to meet the feed water. Dry steam is drawn from the steam drum; the water line being at about the middle of the steam and water drums.

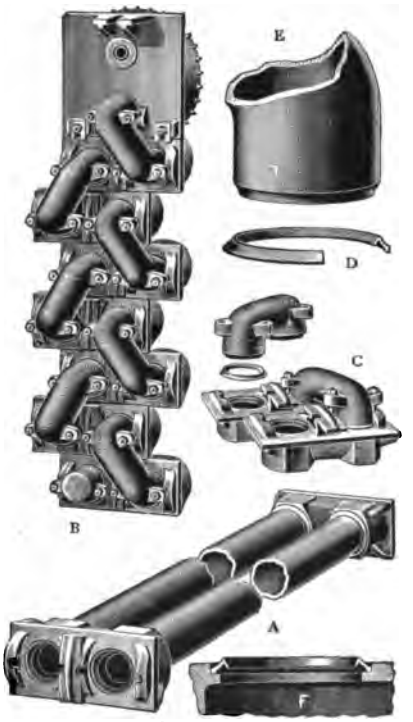


Fig. 23.

The method of conducting the gases from the fire is practically the same as in the Babcock and Wilcox boiler. They pass among the tubes three times before escaping to the chimney.

Numerous sliding doors in the brick work give access to the interior for repairs, cleaning, and inspection. A steam jet is used to clean the soot and dirt from the tubes. The efficient mud drum makes it necessary to clean the inside of the tubes only at long intervals.

The prominent features of this boiler are:

Safety. Being a true sectional boiler, even to the separate water drums.

Flexibility. The pairs having flexible connections and freedom for expansion.

Almy Boiler. Fig. 24.

Dry steam. From the use of the separate steam drum.

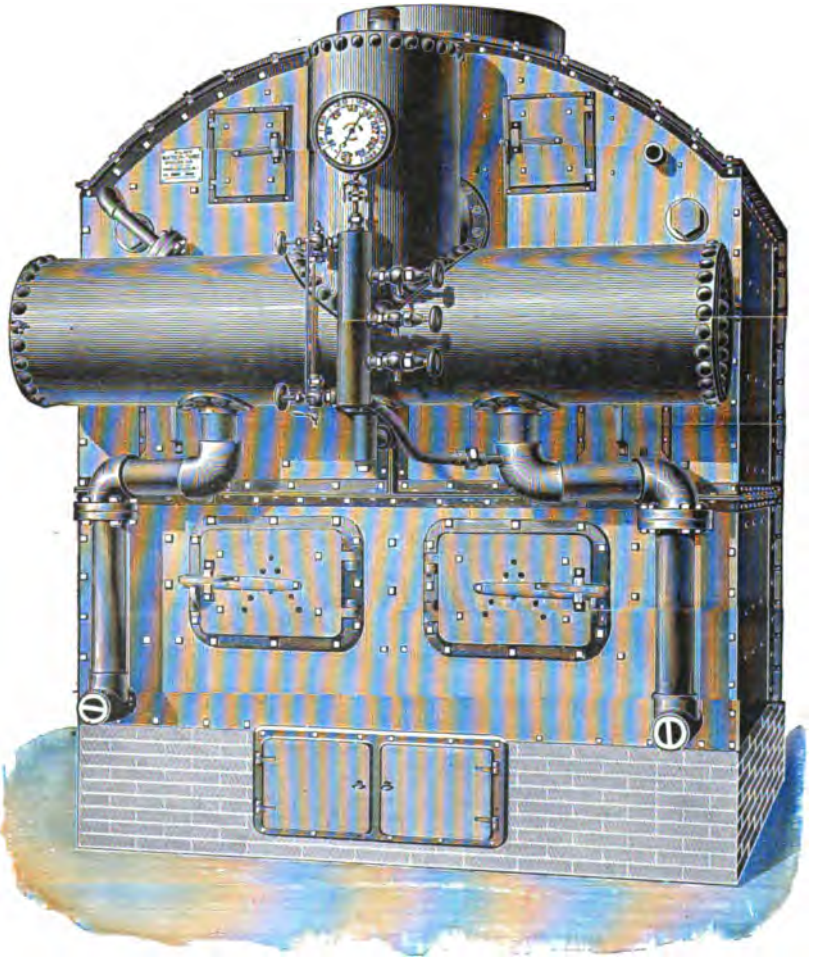
Ease of repair. Any of its parts can be replaced in a short time and without the aid of an experienced boiler maker.

Increase of capacity. More sections can be added with ease, either on top or at the sides of those already in place.

Ease of setting up. It can be put into a building through an opening forty inches square.

It is said to make dry steam without any fluctuations of pressure, and to be an economical and rapid steam generator.

The Almy Sectional Water-tube Boiler has the general prin-



Almy Boiler. Fig. 25.

ciples of all water-tube boilers, and is used for both marine and stationary work. The general arrangement of tubes is shown in Fig. 24; the fire-box and casing being removed. The base is a large pipe called the manifold, which extends along each side and

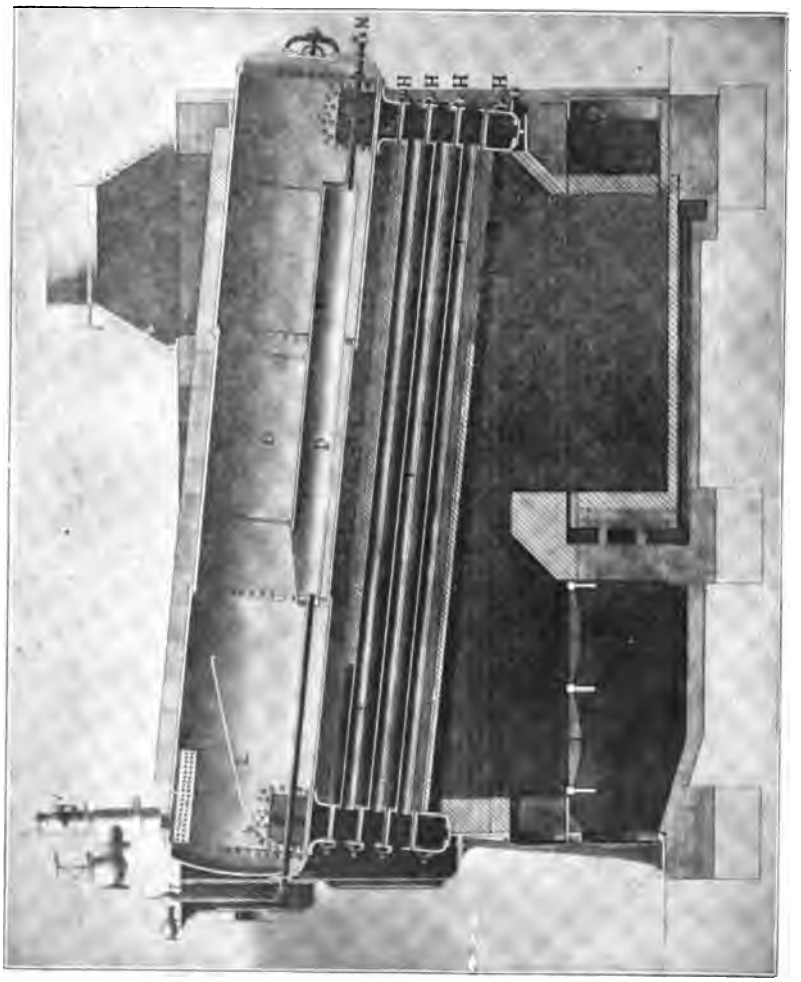
across the rear. At the top a similar manifold extends along the sides and across the front. The sections at the sides rise vertically to a height sufficient for the crown of the fire-box, and returning to the side, connect by vertical tubes with the top manifold. The rear sections rise from the manifold at the back, extend at right angles above the tubes from the side sections, then return back, then to the front again and connect by vertical tubes with the top manifold. A vertical separator connects the top manifold with a horizontal reservoir at the front, as shown in Fig. 25, which is a front view of the boiler showing casing, doors, gauges, etc.

The feed water from the top heater enters the boiler at the bottom of the horizontal water reservoir. It flows through the downward flow pipes to the lower manifold, thence up through the generating pipes. The water is separated from the steam in the separator and falls to the water reservoir.

This boiler is a quick steamer and is used with a duplex pump for putting out fires. It is so constructed that any section may be stopped off and repaired when convenient, or the section may be removed altogether without even disturbing the casing. It may be taken apart and transported in packages of less than 300 pounds each.

The Heine boiler is a water-tube but not a sectional boiler. In principle, it is similar to the Babcock and Wilcox, but differs in detail. The tubes are inclined about one inch in twelve and the steam and water drum is parallel to the tubes. Fig. 26 is a sectional view, showing the boiler complete in its brick setting.

The tubes are of wrought iron and extend between, and are expanded in the inside faces of the water legs. The external appearance of one of the water legs, which is composed of a parallel head and tube plate, flanged and riveted to a butt strap is shown in Fig. 27. These two plates are strengthened by short hollow stay bolts. The water leg is curved at the top and joined to the shell by flanged and riveted joints. The heads of the steam and water drum are dished or spherical and require no staying. Opposite the end of each tube and in the head plate, is placed a hand hole, which is slightly larger than the tube and through which it can pass.



Heine Boiler. Fig. 26.

The products of combustion pass over the bridge wall to the combustion chamber, which is large enough to insure complete combustion. The hot gases from this chamber, guided by tiers of tiling, pass along the tubes nearly to the front, where another tier of tiling compels them to go toward the back of the boiler under the shell and around it to the uptake. The boiler is supported on the brickwork at the two headers.

The feed water, from the heater, enters the drum through the pipe F, and flows in to the plate iron mud drum D, where it passes to the rear with greatly lessened velocity. As the water is warm and moves slowly, the sediment is deposited on the bottom of the mud drum, the removal of the latter being provided for by the blow-off pipe N. The water becomes hot in the mud drum, rises and flows in a reverse direction toward the open front of the mud drum, where it passes in a thin sheet to the rear of the steam drum; it then enters the water leg and flows from there to the tubes. During its flow through the tubes, it is partially converted into steam, which rises easily through the thin layer of water in the front end of the drum. A deflection plate L prevents water being carried to the perforated steam pipe A.

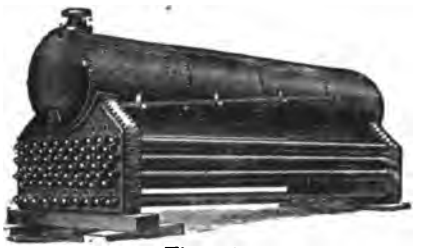


Fig. 27.

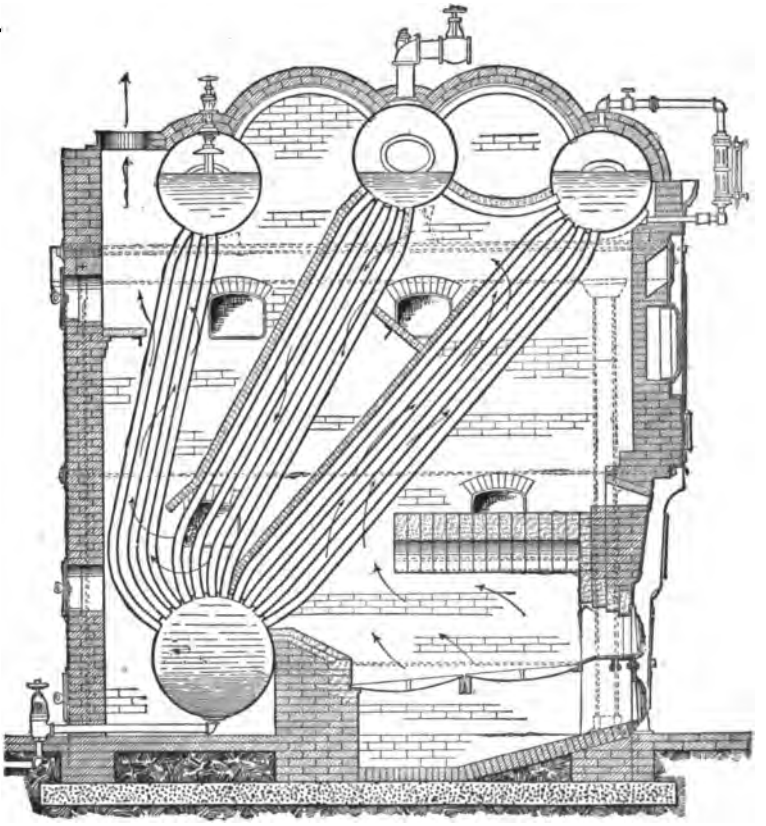
The following are some of the advantages claimed for the Heine boilers:

Economy and capacity; it readily absorbs and transmits heat; its circulation is swift and unimpeded and there are no contracted parts. It is designed to have slight variation of temperature and also an equalized expansion. The steam is dry, because the direction of the currents of water is away from the steam pipe. It requires little cleaning as the scale and mud are precipitated while the feed water moves very slowly in the mud drum, which is away from the fire so that no sediment will be baked on the plates.

The Stirling Boiler. This form is a water tube but not a sectional boiler. It consists of three cylindrical steam and water drums at the top and a large drum at the bottom, connected by

three sets of steam circulating tubes slightly curved at the ends. The steam spaces of all the upper drums are connected by pipes, and the front and middle drums are connected by water circulating tubes.

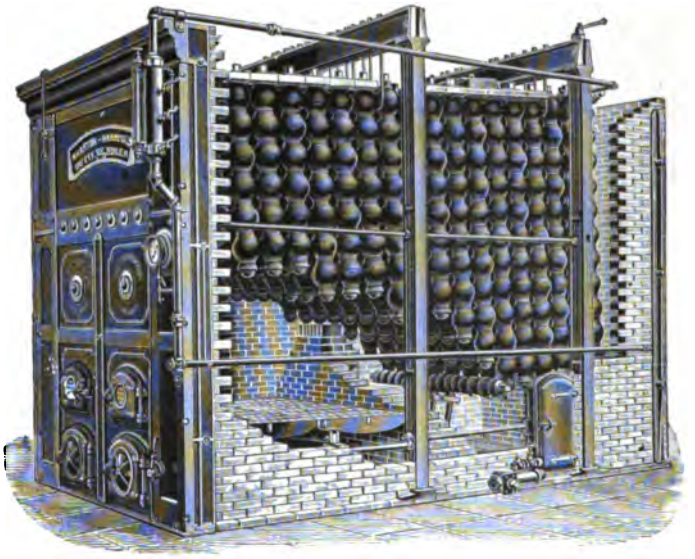
The feed water enters the rear upper drum and coming in



Stirling Boiler. Fig. 28.

contact with the gases just before they enter the uptake, becomes gradually heated. This heating causes most of the sediment to fall to the mud drum while the water descends. The sediment in the form of mud may be blown off at intervals from the mud drum. This mud drum is protected from the intense heat of the furnace by a bridge wall, and is an excellent settling chamber.

Fig. 28 shows the manner of separating each set of tubes from the others by partition walls of fire-brick tile, so that the gases from the furnace are directed as shown. These gases rise from the furnace, pass among the first set of tubes, are then guided downward through the second set, and after rising again among those of the third set, escape to the chimney. As the gases are compelled to remain in contact with the tubes for a considerable period, they escape at a very low temperature. Thus a



Harrison Safety Boiler. Fig. 29.

large portion of the heat is utilized. The fire-brick arch just above the furnace insures an even distribution of the gases and promotes combustion. This arch heats the entering air to a high temperature, thus reducing the liability of chilling the metal by an inrush of cold air.

It is claimed that this boiler is a safety boiler because all parts are of wrought metal and are either cylindrical or spherical in shape, thus reducing the tendency to distortion. The curved tubes reduce the strains due to unequal expansion and contraction.

Steam is taken from the middle drum which is set a little higher than the others in order to obtain more steam space and dry

steam. There are numerous openings in the brickwork, which allow entrance for cleaning.

This boiler is adapted for use in cramped spaces, since it can be made broad with little height, or high with small floor area.

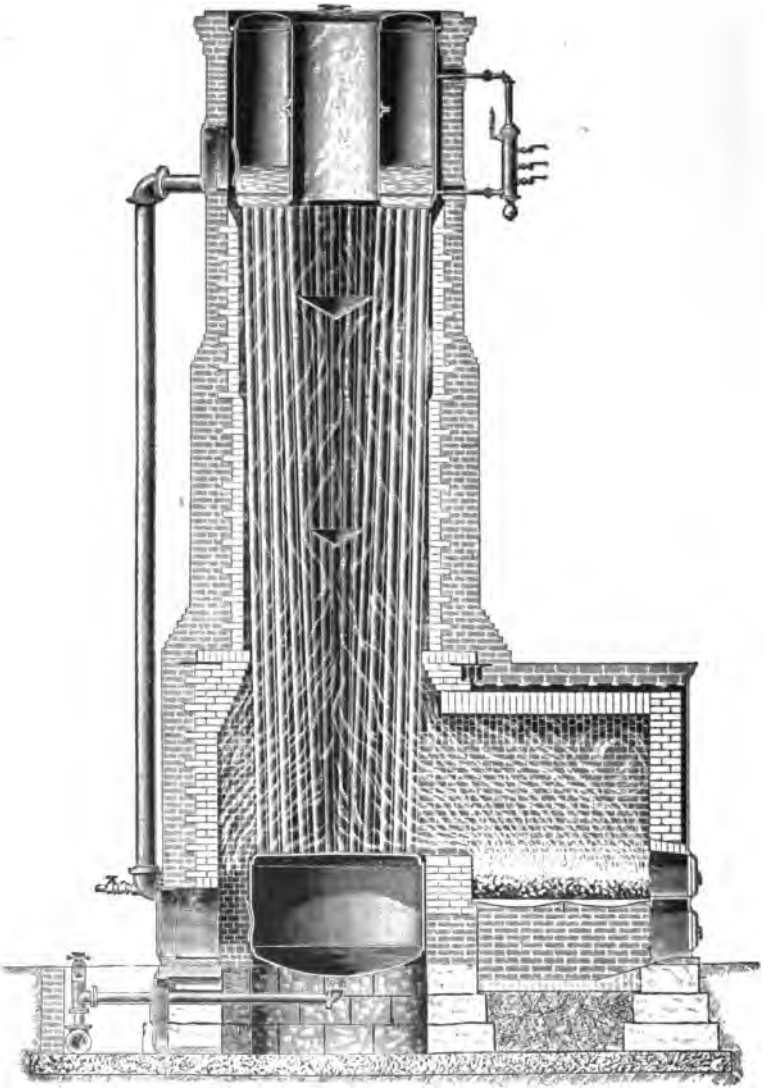
The Harrison Safety Boiler. We now consider a boiler shown in Fig 29, which differs in many respects from those already described. It belongs to the class of water-tube sectional boilers; but in place of tubes, in which the water circulates, hollow cast iron spheres are used. These spheres are called "units," Fig. 30, and are arranged in vertical sections called slabs, which are suspended side by side, about one inch apart, from an iron framework. The brickwork setting is merely a covering and does not support

Fig. 30.

the boiler, and can be set or repaired without disturbing the same.

The use of units in place of tubes, combines great strength and large heating surface. They are strong because spherical, and on account of the division of water into small portions, the heating surface is large and efficient. The units are held together by bolts which pass through the centers. The joints are made steam and water tight by means of machined faces without the use of any packing.

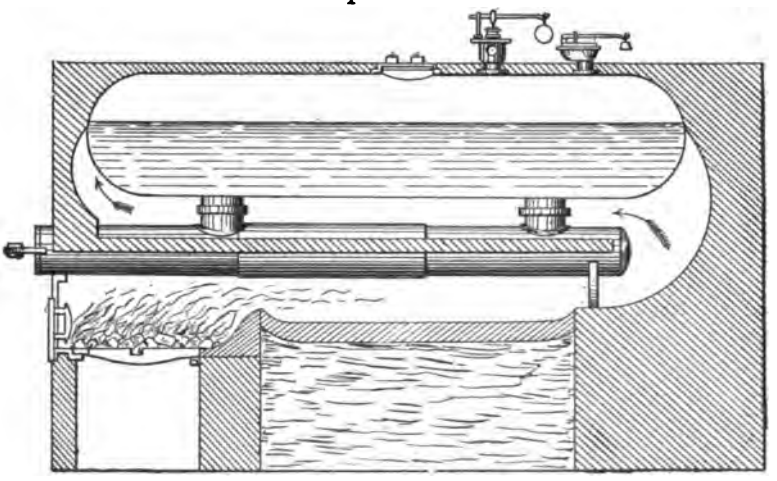
The fittings are similar to those of other boilers.



Cahall Boiler. Fig. 31.

Safety is the great advantage of this form of boiler. From the method of joining, rupture cannot extend beyond the unit, which prevents disastrous results. They are claimed to be durable, economical, rapid steamers, furnishing dry steam, and easily set up, cleaned, repaired and transported. The capacity can be increased by merely adding more slabs.

The Cahall Boiler is a vertical water tube boiler, shown in section by Fig. 31. The steam and water drum is annular and about one-third full of water. The cylindrical mud drum at the bottom is connected with the steam and water drum by tubes within the brick setting and also by two large circulating pipes outside the brick setting. The fire is in a brick furnace at one side of the boiler. The hot gases pass among the tubes back and forth, to and from the central space between the tubes. The baffle plates of triangular section, placed in the central space, break up the currents of gases. The water circulation is downward from the top by the outside pipes to the cylindrical drum, then upward through the water tubes to the steam and water drum. Steam is drawn from a nozzle at the top.



Elephant or French Boiler. Fig. 32.

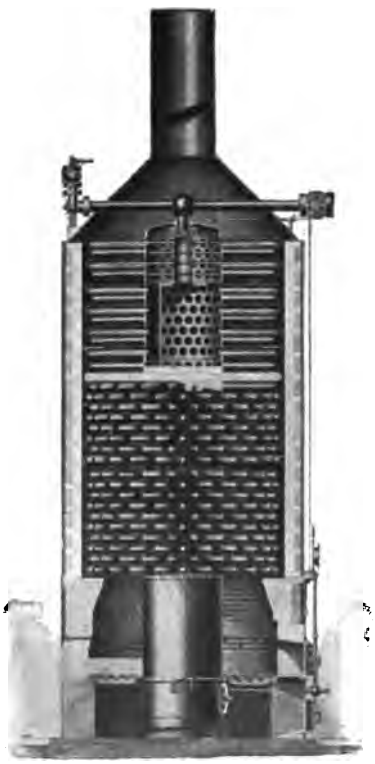
Both drums may be entered by manholes and eight handholes in the upper drum facilitate the removing of any tube without disturbing the others.

The Elephant or French Boiler is seldom used in England

or America. Its general arrangement is shown in Fig. 32. The large cylindrical shell with hemispherical ends forms the steam and water drum. It is connected to the two or three smaller cylinders or heaters by vertical tubes or legs. Brick settings support the boiler and guide the products of combustion.

The hot gases pass to the rear under the heaters, which are separated from the steam drum by brick work, as shown in the engraving, thence through the side flue to the front, returning to the rear under the large drum and finally reaching the chimney by way of another side flue.

The heating surface is much greater than that of the plain cylindrical boiler and the large cylinder or steam drum is removed from the intense heat of the fire. By this arrangement, the cylinder is not exposed to the changes of temperature occasioned by the opening of the furnace door. The small cylinders are strong and can be replaced without much difficulty.



Hazleton or Porcupine Boiler. Fig. 33.

The Hazelton or Porcupine Boiler, Fig. 33, has a peculiar form. It is a vertical water tube boiler, but unlike the Cahall boiler, the tubes are short and horizontal. The grate is circular and formed around a central stand pipe, which rests on a circular cast iron foundation. This stand pipe, which is about three feet in diameter for large boilers, is used above the grate as a heating surface and steam drum, while below the grate it forms the mud drum, which may be entered by a manhole just below the base. Above the fire the stand pipe is provided with radial horizontal tubes, from whose appearance the boiler derives its name. They are about four inches in diameter and two and one-half feet long,

for a large boiler, with the outer end closed and hemispherical. The inner end is expanded into the plate of the stand pipe

The tubes which extend nearly to the brickwork, are free to expand and contract.

By means of short pipes, the steam is dried before entering the supply pipe, thus preventing water from being carried over to the engine.

MARINE BOILERS.

In the earliest forms of marine boiler, working with a pressure of about thirty pounds per square inch, the boilers were made square or box shaped. They were economical and of convenient form for ships. But when higher steam pressures were demanded, the flat surfaces required so much staying, to retain their shape, that they were abandoned and the cylindrical type introduced as that is the most practical shape to resist internal pressure. The cylindrical boiler is not as economical as the box-shaped boiler, but it will withstand very high pressures. At present, the Scotch or drum boiler is the most extensively used, though many forms of water-tube boilers appear to be adapted to this work and as higher pressures are needed it is probable that this type will be used instead of the drum boiler, on account of the decrease in weight.

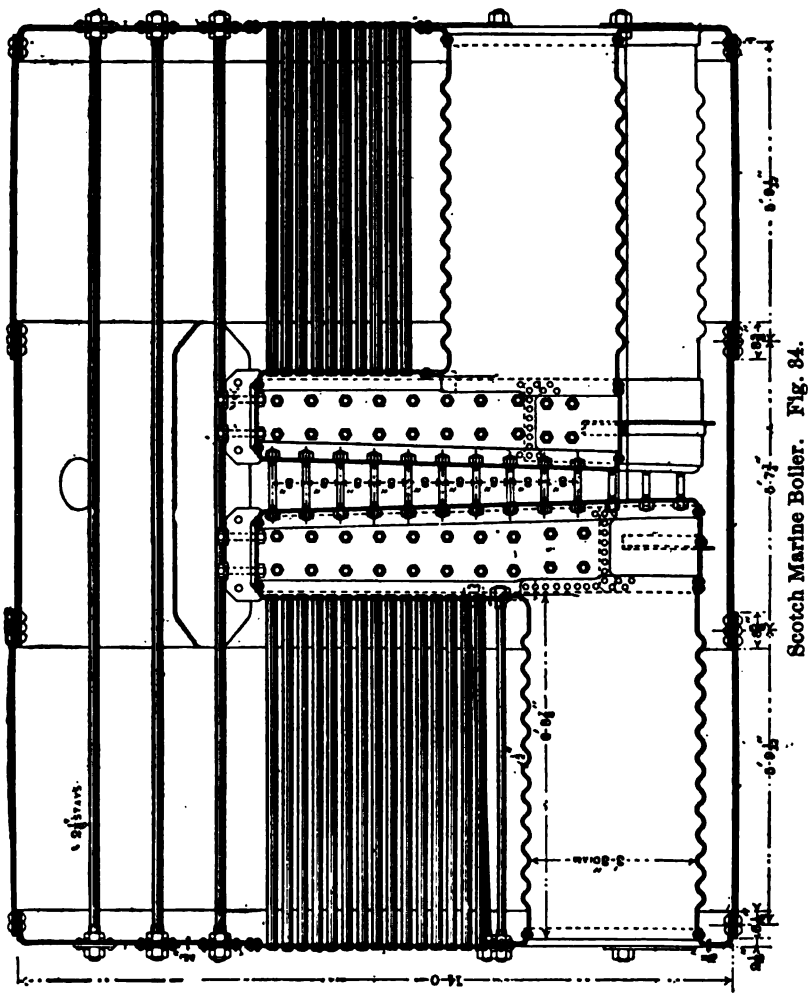
The two common types of fire tube marine boilers are the *Return* tube and the *Through* tube.

Let us consider the single-ended Return tube Boiler. This boiler combines the internal flues of the Lancashire and the small fire tubes of the multitubular. It has a cylindrical shell made of plates riveted together, with flat ends which are flanged and riveted to the shell. The boilers are from 7 to 11 feet long and from 10 to 15 feet or more in diameter.

The furnaces are in the internal flues which are cylindrical and from 3 to 4½ feet in diameter. There may be two, three or four furnaces, the one represented by Figs. 34 and 35 (a double-ended boiler) having three in each end. Several methods are used to strengthen the flues; the corrugated tube being the most successful. It is made of one piece as shown in section in Fig. 36. It is strong to resist external pressure, has a large heating surface and the

curved portions take up the expansion. Other methods have strengthening rings and expansion joints, the flue being made in sections as in the Cornish boiler.

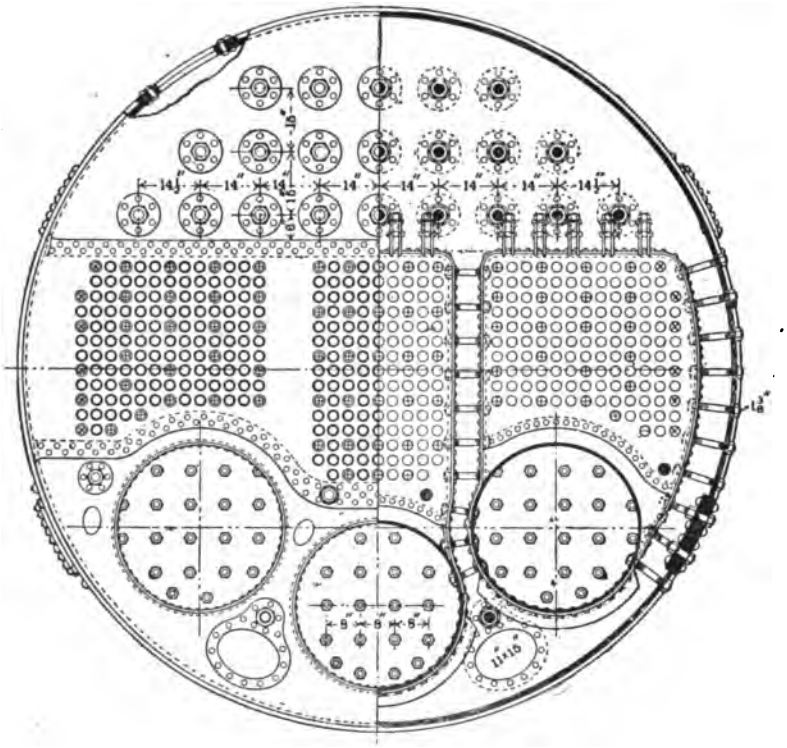
The rear ends of the furnace flues open into the combustion



Scotch Marine Boiler. Fig. 84.

chamber, in which the gases from the furnaces are completely burned. Each flue may have its combustion chamber or two flues and more may open into the same one. A three flue boiler usually has three combustion chambers, while a four furnace boiler has

two, into each one of which two flues lead. Fire tubes connect the front plate of the combustion chamber with the front head of the boiler, and are made small to provide a large heating surface



Transverse Section Scotch Marine Boiler. Fig. 35.

within a small space. The grate bars divide the flue into two parts, the top being occupied by the fire and the lower part by the ash

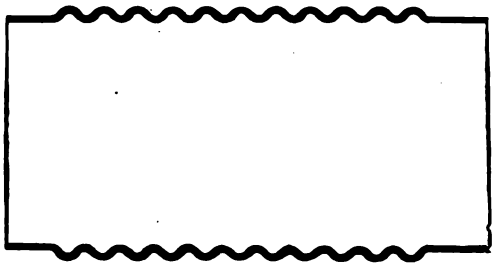
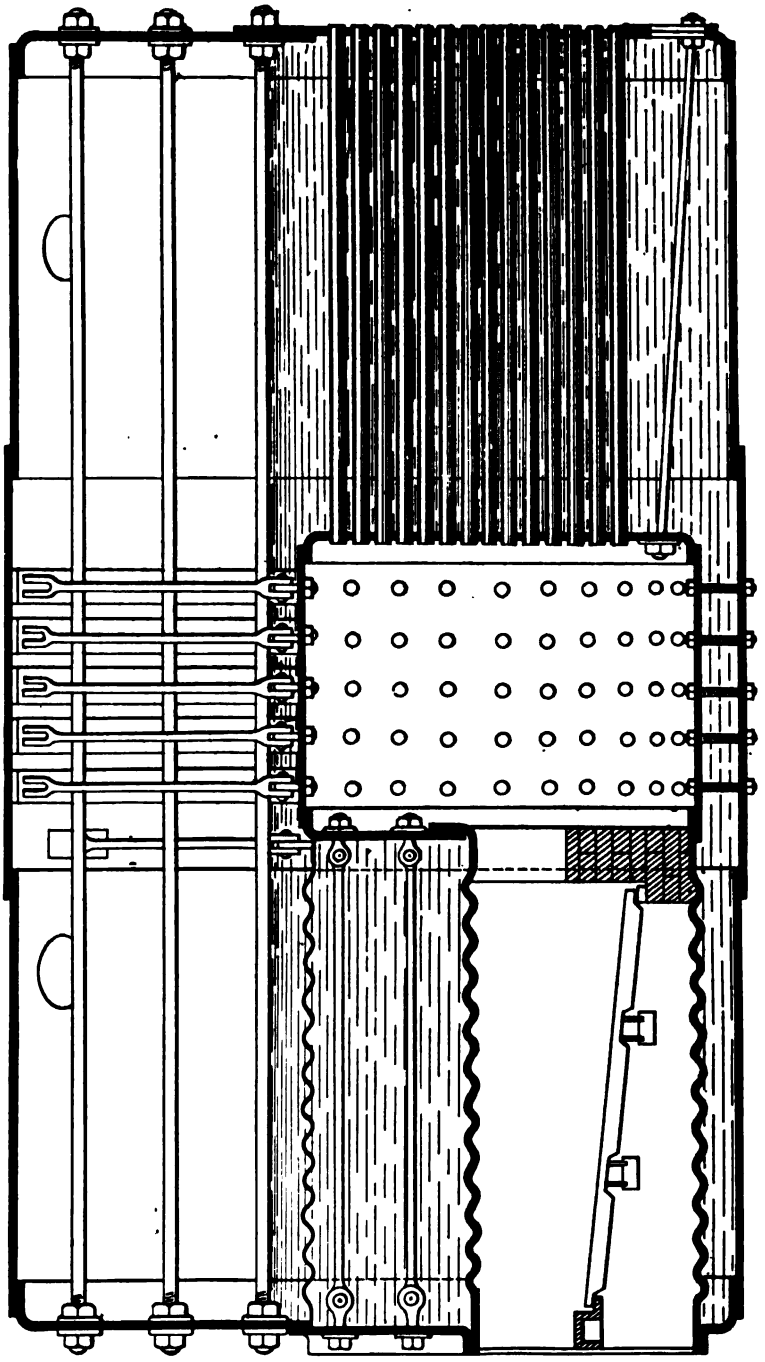


Fig. 36.

pit. The backs of the furnaces are formed of fire brick bridges. The steam space is the portion above the water line as in a multi-



Through Tube Marine Boiler. Fig. 87.

tubular boiler. The heating surface is composed of the internal surface of the flues and tubes, and the plates of the combustion chamber. The uptake is situated at the front end of the boiler.

The flat ends require staying to prevent them from bulging, which is done by through stay rods in the steam space and by making some of the tubes thicker than the others and screwing them into the tube sheet. The flat plates of the combustion chamber are stayed to the rear end by short screw stay bolts with nuts and washers, in place of being riveted over as in locomotive work.

The hot gases from the furnace enter the combustion chamber where they are completely burned; the products passing through the tubes to the uptake and from there to the funnels.

In order to save the weight of the back plates and to prevent loss of heat by radiation, the double ended boilers are used. In this construction the furnaces are placed in each end of the boiler, the combustion chambers being near the center, as shown in Fig. 34. Another form has a common combustion chamber for each opposite pair of furnaces. The gases from the two furnaces mix in the common combustion chamber and then pass through the tubes to the opposite ends of the boiler. In other details the double ended boilers are the same as single.

The circulation is defective in the Scotch or drum boiler, both when single and when double ended, and it takes several hours to raise steam.

The Through Tube Marine Boiler, a longitudinal section of which is shown in Fig. 37, is used on ships with small draft of water, because they can be made of small diameter. The various parts of this type of boiler are the same as those of the return tube marine boiler, but they are arranged differently. The rear plate of the conbustion chamber forms one of the tube plates, and the end of the boiler the other tube plate. The top of the combustion chamber is stayed to the shell by sling stays. On account of the tubes not being over the furnace, this boiler must be long if it is to evaporate considerable water. Fig. 38 is a half section and end view of this boiler, which shows the arrangement of the furnace and tubes.

Water tube boilers are used in marine work when it is

desired to have a large power for the weight and when high pressures are used. Those types that do not require a brick setting are well adapted for this purpose.

The Almy Boiler, already briefly described with land boilers, is used to some extent for general marine work.

The Belleville Boiler, Fig. 39, consists of a series of tube coils with bends and elbows. At the top, there is a steam drum

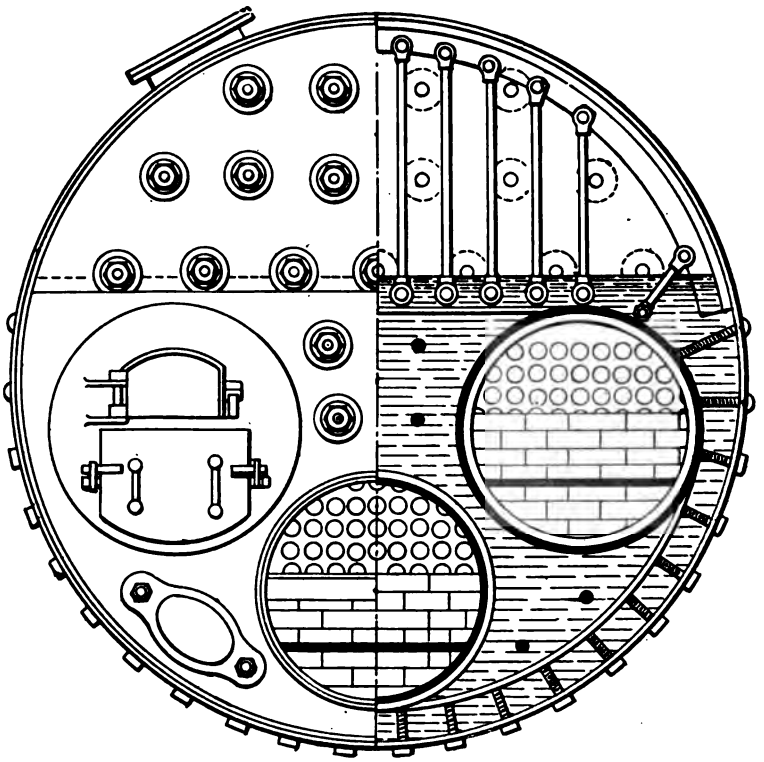
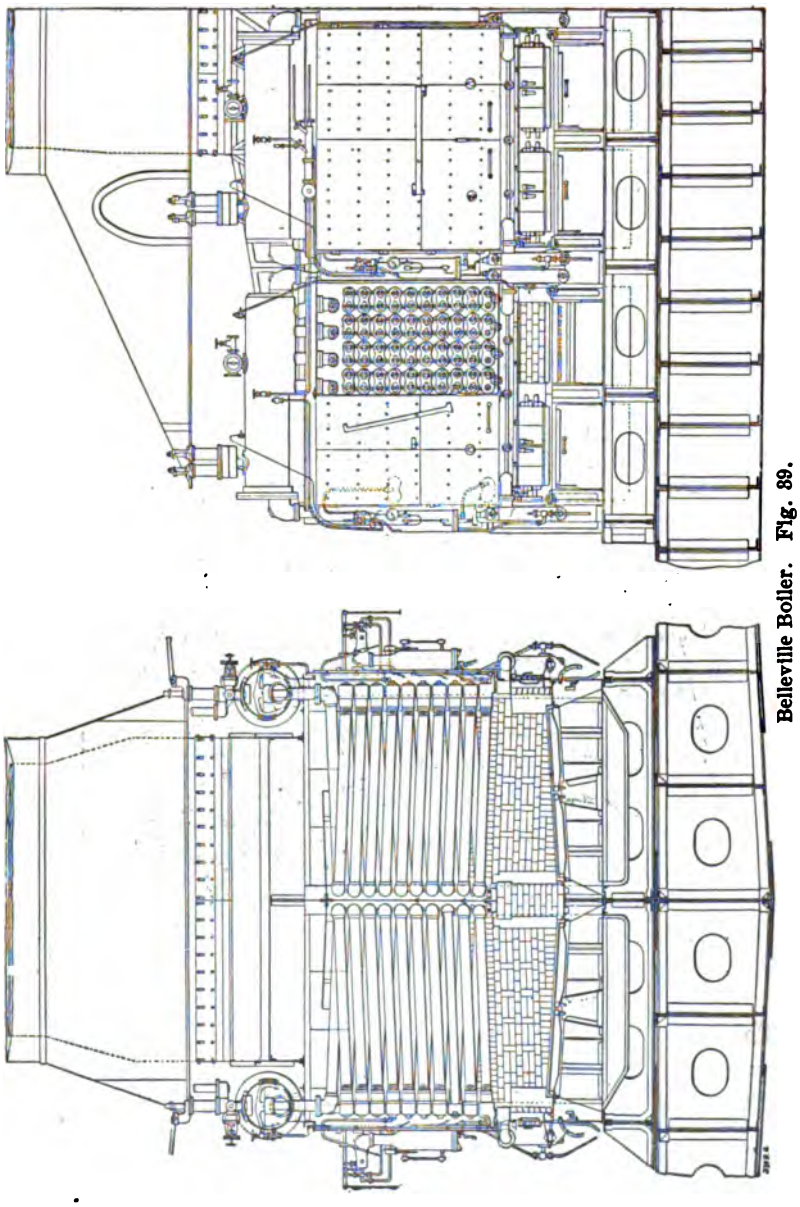


Fig. 38.

which is connected with the mud drum at the bottom by two circulating pipes. All the coils or elements are connected at the bottom to the rectangular feed pipe, each one being continuous to the steam drum.

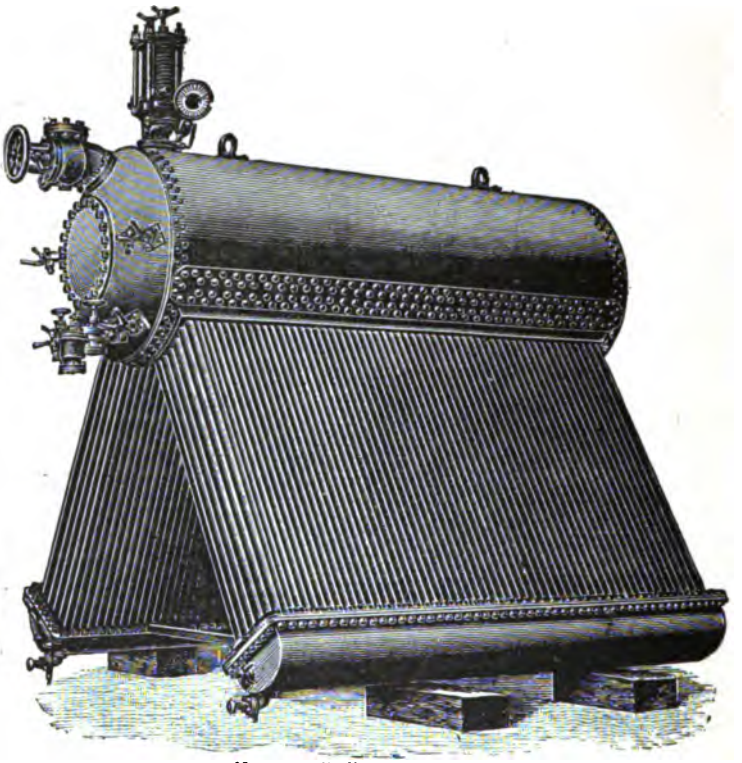
The amount of water in the boiler is very small, and steam may be raised very rapidly on this account, but an automatic feed regulator is necessary. A mixture of steam and water is delivered



Belleville Boiler. Fig. 89.

from the elements to the steam drum which is not an efficient separator, so an external separator is placed between the boiler and the engine.

The circulation of water is from the entrance in the steam drum, downward through the circulating pipes to the mud drum, where the sediment is deposited, thence up through the elements to the steam drum.

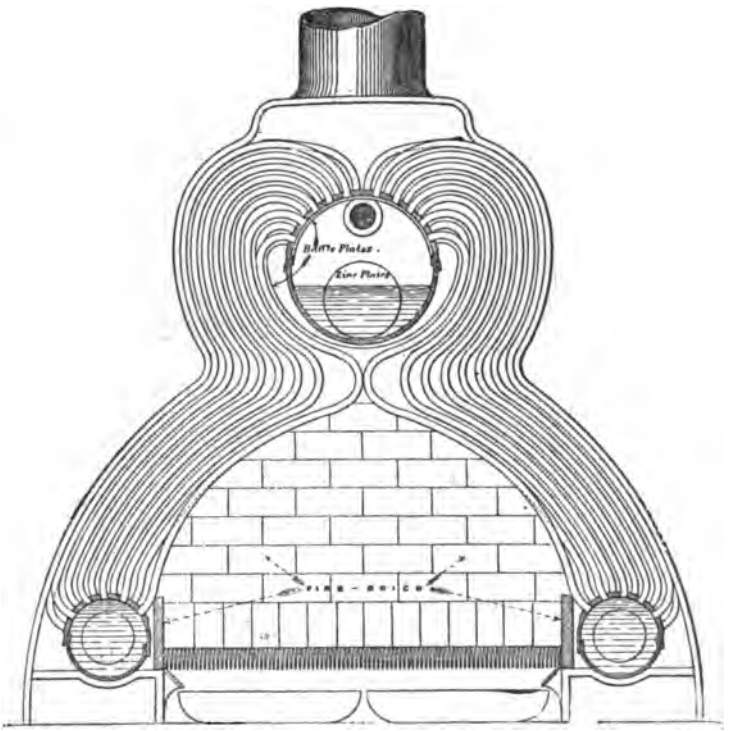


Yarrow Boiler. Fig. 40.

The Yarrow Boiler is used principally for torpedo boats. There are three drums, two lower or mud drums and one upper or steam and water drum. These drums are connected by straight tubes, as shown in Fig. 40. Some of the tubes carry the cool water from the steam drum to the mud drum and others carry water and steam to the upper drum. Those receiving the most heat from the fire and making the most steam deliver to the upper drum.

This boiler, like the Almy, is enclosed in an iron casing, which prevents the escape of heat to other parts of the vessel. The heat radiated from the casing is to a great extent carried back to the fire since the air for combustion is drawn from an iron box which envelopes the casing.

The Thornycroft Boiler, represented by Fig. 41 is used on torpedo boats, steam launches and yachts, and is named from its designer, Mr. Thornycroft. The boiler consists of a large steam drum or separator connected by numerous bent water tubes with two or three smaller drums at the bottom. The large drum at the

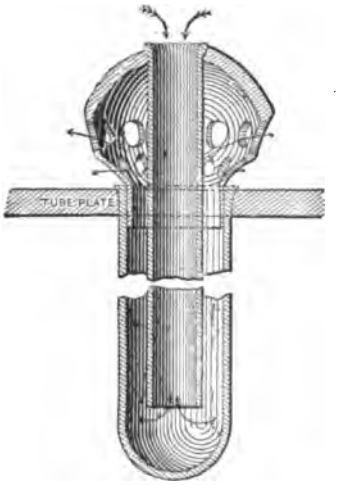


Thornycroft Boiler. Fig. 41.

top is connected to the other drums by small pipes. The setting is an iron frame, and the whole is encased in a thin iron shell lined with a non-conducting material. The boiler may be cleaned by means of numerous doors in the casing.

The circulation is downward from the top drum to the side drums and upward through the bent tubes to the steam drum. A baffle plate extends above the water line in front of the tubes. Steam is drawn from the top drum by a nozzle at the rear.

The products of combustion pass among the tubes that lead from the side drums. The outer tubes take the heat that goes toward the casing. This boiler is successful with forced draft. It makes steam freely and rapidly and gives little trouble, but like the Yarrow boiler the small amount of water used makes it necessary to regulate the feed supply carefully.

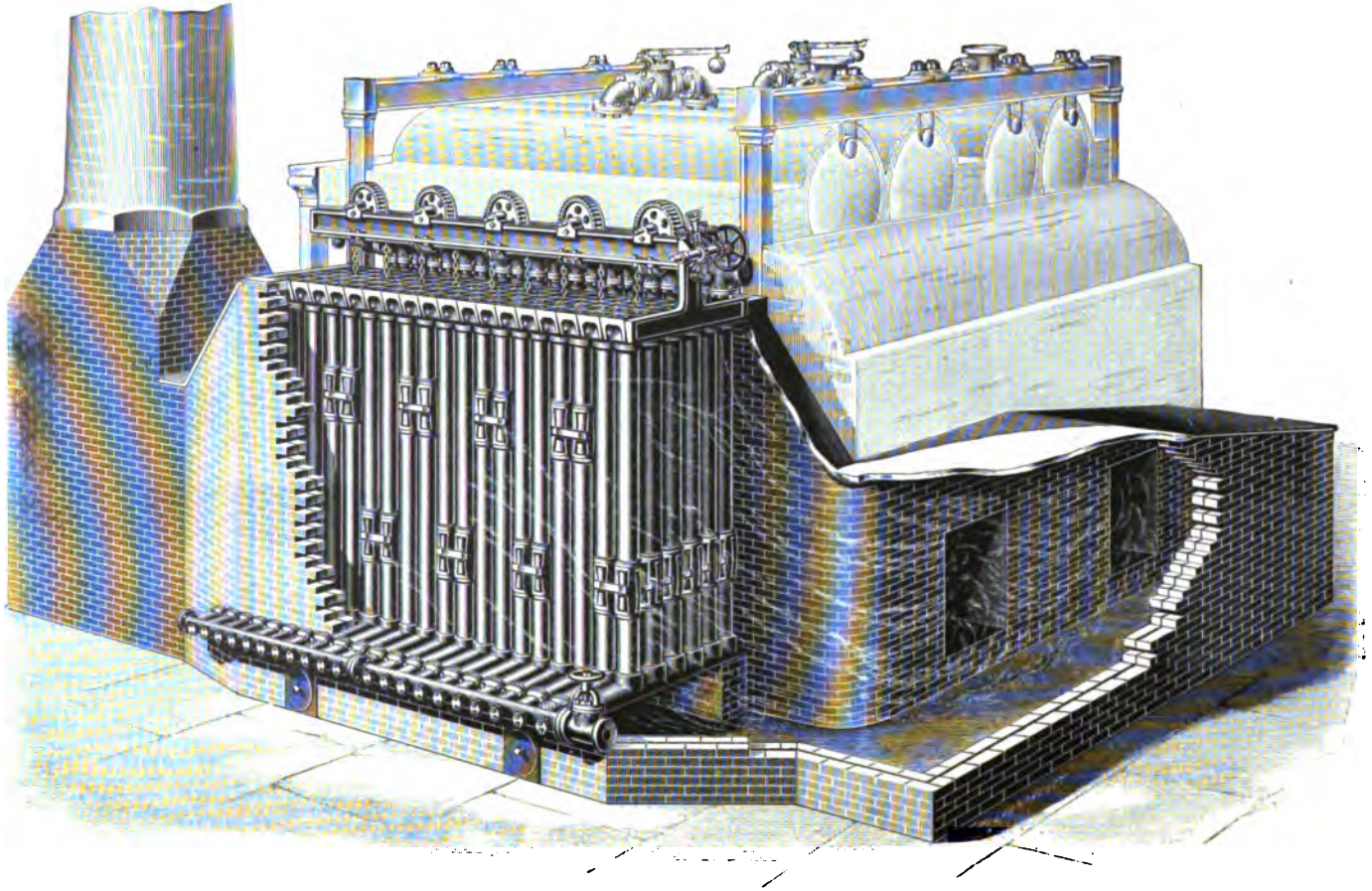


Field Tubes. Fig. 42.

The Field Boiler has a vertical cylindrical shell in which there is a fire box with a flat top. A flue passes from the crown sheet through the head to the chimney. The water tubes, called Field tubes, hang downward from the crown sheet into the furnace. Fig. 42 shows a section of one of these tubes. The outer tube is expanded into the crown sheet at the upper end; the lower end being closed. The inner tube, which is suspended by a perforated internal support, is open at both ends and is suspended inside the outer tube. The upper end of the inner tube is expanded to give free entrance to the water.

The circulation is downward through the inner tube, thence upward through the outer tube. On account of the thin layer of water exposed to the action of the fire, steam may be raised very quickly. This form of tube is used in fire engine boilers.

If the tubes hang downward, they are called drop tubes and have an internal tube as shown or a partition. When it is used horizontally, the internal tube is supplied with water from a compartment separated from that to which the external tube is attached. Boilers for steam heating are sometimes supplied with these drop tubes.



GREEN FUEL ECONOMIZER

SECTION 5.

BOILER ACCESSORIES.

FURNACES.

The furnace varies in shape, size and detail with the type of boiler and the kind of fuel. It consists of a mouth piece with doors, grate bars, bridge and ash-pit; some furnaces have special arrangements for the removal of ashes, clinkers, etc., for cleaning and for smoke prevention. There are some general conditions that every furnace should have, in order that the highest efficiency be attained. There should be a uniform and abundant supply of air to the under side of the grate. This is easily obtained in most furnaces; exceptions being the Cornish, Lancashire, Return Tube and Scotch Marine boilers, since they are internally fired.

When smoky fuels are used a small amount of air is necessary on the surface of the coal, for a few minutes after firing. This air is generally supplied through a grid or draft plate in the fire door. As the air thus admitted is usually cold, the amount must be small to prevent cooling of the furnace and checking of combustion.

In order to prevent radiation, many boilers have a large casting for a furnace front. In the case of an internally fired boiler, this casting conceals the rings of rivets which join the flues to the end plates. A mass of brickwork at the front of the boiler is likely to keep moisture in contact with the end plate, and conceal outward bulging, due to defective stays or plates; and is likely to cause corrosion. A coating of non-conducting material which can easily be removed is preferable. The furnace should be so arranged as to allow **Complete Combustion.** In an externally fired boiler, this is done by making the space above the fire sufficient to allow the gases to become completely burned. The average distance between the grate bars and crown sheet is usually about two feet. If the distance is increased, some of the effect of the heat is lost; if the distance is small, the plates are likely to be damaged, and the combustion impaired. In the internally fired

boiler, the combustion and radiating spaces are often sacrificed for a large grate area. If the bars are placed too high, the combustion chamber must be used in order to insure complete combustion.

The locomotive crown sheet is from four to six feet above the grate bars.

The **Furnace Door** is usually of cast iron and is supplied with a circular or sliding draft plate which admits air to the top of the fire. It is usually protected by a perforated wrought iron plate, bolted to the casting, an air space of two or three inches being allowed.

Grate. Grates should not be larger than the fireman can conveniently clean and fire. A narrow grate should not be as long as a wide one, and if it is more than four feet wide two furnace doors should be used. The grates are often inclined slightly to aid in firing and in order that the space above and below may be used to the best advantage.

The **Grate Bars** are usually made of cast iron, as this material is cheap and in most instances lasts as well as wrought iron. They are made in various forms, according to the fuel and shape of the

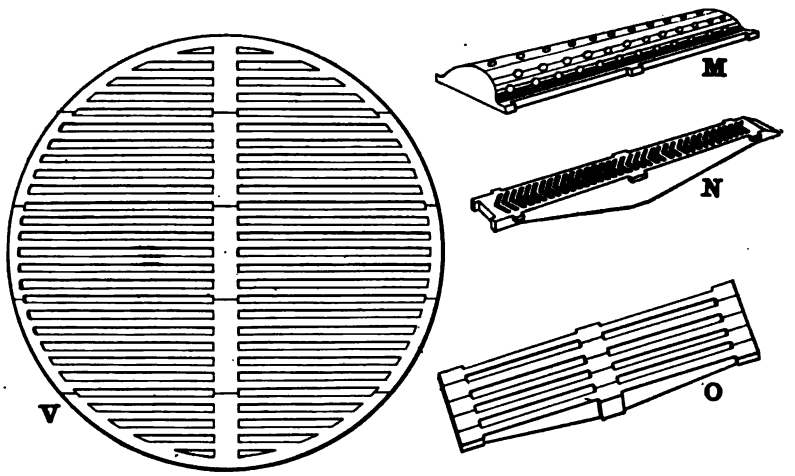


Fig. 1.

fire-box. Fig. 1 shows a few common shapes. The one shown at V is for a circular grate in a vertical boiler, N is called the herring-bone grate, M, the form used for burning wood and saw-dust, and O is the ordinary form.

If fine coal is used, the perforations, or air spaces, must be small; the area varying from 30 to 50 per cent. of the total area of the grate. The grate is usually made up of parallel bars and should *never* be made in one solid casting, as it is likely to break from expansion strains. The bars should not be more than three feet long and should be thinner at the bottom than at the top. In order that the spaces may remain the same, even after the top surfaces have become worn, they are made the same thickness for about $\frac{3}{4}$ of an inch from the top. Bars having a section like those shown in Fig. 2 give good results. They are about three inches deep in the middle and have distance pieces at the center as at the ends, to prevent twisting. The air spaces are about $\frac{1}{2}$ inch wide. If the coal is anthracite, which breaks up, the spaces should be as small as the draft will permit; if bituminous, or a coal that cakes considerably, the spaces may be made larger.

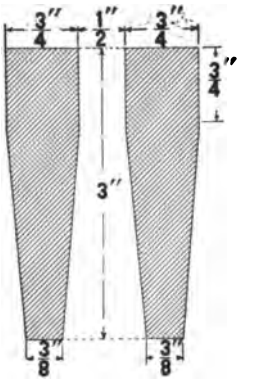


Fig. 2.

In order to facilitate handling, they are often cast in sets of two or more, and when set should be allowed space for expansion at each end, the amount being about one inch in twenty-four. Wrought iron bars are used in locomotives and in boilers where the bars are subject to hard usage. The point of fusion of wrought iron is higher than that of cast iron; but wrought iron bars bend and twist more than bars of cast iron.

Bars are sometimes hollowed and currents of water made to flow through them. By this method their durability is increased and also the efficiency of the furnace. The extra cost, however, and the expense of keeping them in order usually offsets the above advantages.

Rocking Grates. The labor of breaking up clinkers is considerable when ordinary fixed grate bars are used. There are many forms of rocking or shaking grates that do this work without the need of opening the furnace door.

In locomotives rocking grates are necessary, since the rate of combustion is high and the fire must be in good condition at all

times. The grate, being below the cab floor, would be very hard to clean if ordinary fixed grate bars were used.

The various rocking grates are operated in many ways; some are flat or rectangular and are moved by shaking backward and forward, others are triangular and are rotated so as to break up the clinkers. The motion is often controlled by cams and levers.

Fig. 3 shows Kelley's standard rocking grate. Each bar is made up of a number of separate leaves, which can be removed

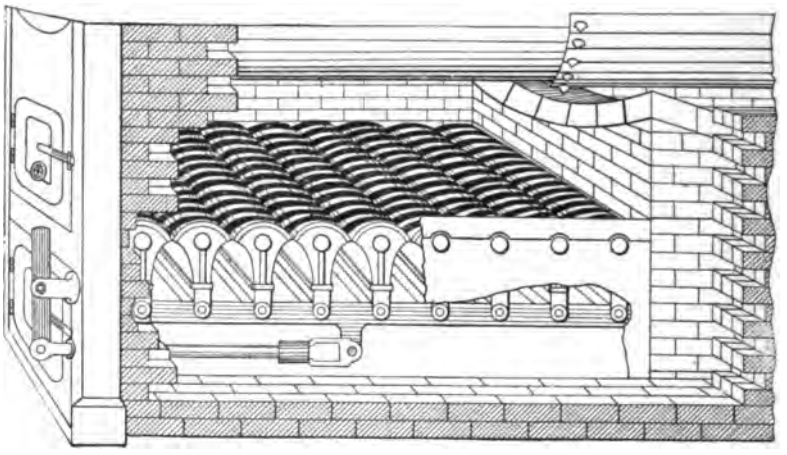


Fig. 3.

and replaced without renewing the whole bar. When the bar is moved back and forth, by means of the lever outside of the brick-work, the leaves rotate through a small angle and grind up the clinkers.

Bridge. The bridge is a low wall or partition at the back of the grate. It is usually made of fire-brick, of cast iron, or of ordinary brick covered with fire-brick. It is sometimes of wrought iron, having a hollow space, which is filled with water in communication with the boiler. The bridge guides the hot gases to contact with the heating surfaces and retards their escape. The height depends on the draft. If the space between the bridge and the boiler is narrow, a considerable draft will be necessary to draw the gases through. The best height can be determined only by trial, as it is dependent upon so many conditions. One rule is to make the space about $\frac{1}{8}$ the area of the grate. Hanging bridges

are often built behind the furnace bridge, the space between being a combustion chamber.

Two or three bridges are often used to keep the hot gases ir contact with the boiler, if it has a long shell.

SMOKE PREVENTION.

In large cities where the escape of considerable quantities of smoke into the air is undesirable, several methods have been devised to prevent the formation of smoke. Among the most important inventions and methods may be mentioned the following: the usé of gas fuel, the introduction of jets of steam to thoroughly mix the air and combustible gases, fire brick arches, mechanical stokers, variation of air supply, down draft furnaces and fuel economizers. *The effectiveness* of the above devices depends largely upon the skill with which they are designed and operated.

The actual loss caused by the escape of smoke, even when it is dense and black, has been found to be slight and usually the appliance used for prevention costs more than is saved. The alternate firing of two furnaces, which open into a common combustion chamber, or the alternate firing of two sides of the same furnace produces a slight gain if the proper amount of air is admitted. But, if in order to burn the smoke, the bed in one furnace, or one side of a furnace becomes thin, there will be no gain in efficiency. The introduction of steam is an efficient method, but it is likely to cause a too rapid rate of combustion. Another arrangement to prevent the escape of smoke is that by which the coal is distilled in a small furnace which is separate from the boiler. The coke and gases thus made are burned in the furnace of the steam boiler. This device is not altogether satisfactory on account of the loss of heat from the detached furnace. Rather than add any smoke prevention device, anthracite or coke is used instead of bituminous coal.

Many engineers and business men consider *a good fireman* to be the best smoke preventer.

DOWN DRAFT FURNACES.

In order to increase economy, capacity, or to prevent smoke.

a down draft furnace is used. In this furnace there are two grates, one a foot or more above the other. Fresh coal is fed to the upper grate, and as it becomes partially consumed falls through to the grate below where the combustion is completed. The draft is *downward* through the upper grate and *upward* through the lower because the connection to the chimney is from the space between the grates. The volatile gases are carried down through the bed on the upper grate and are burned in the space below it where they meet the hot air drawn upward from the lower grate. A large proportion of the air for combustion enters the door at the upper grate. Tests on the Hawley furnace show that 30 to 45 pounds of coal per square foot per hour can be burned with good results.

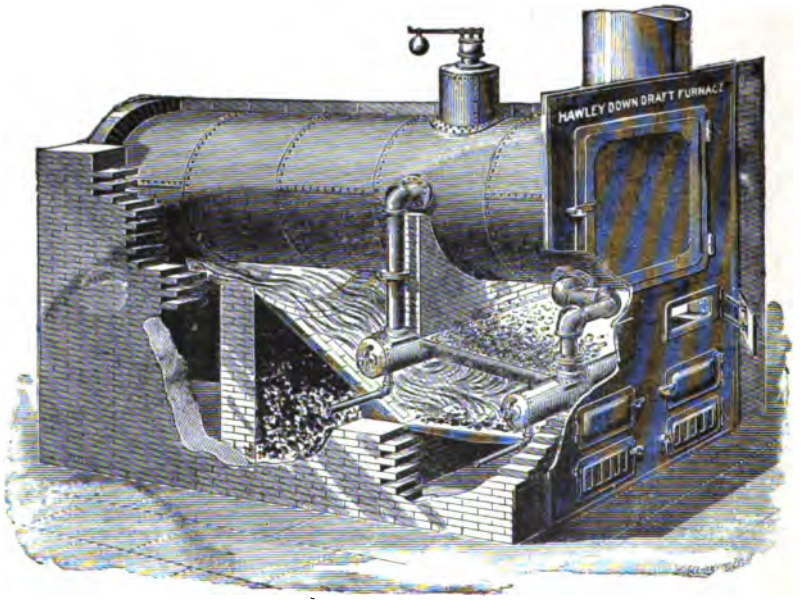
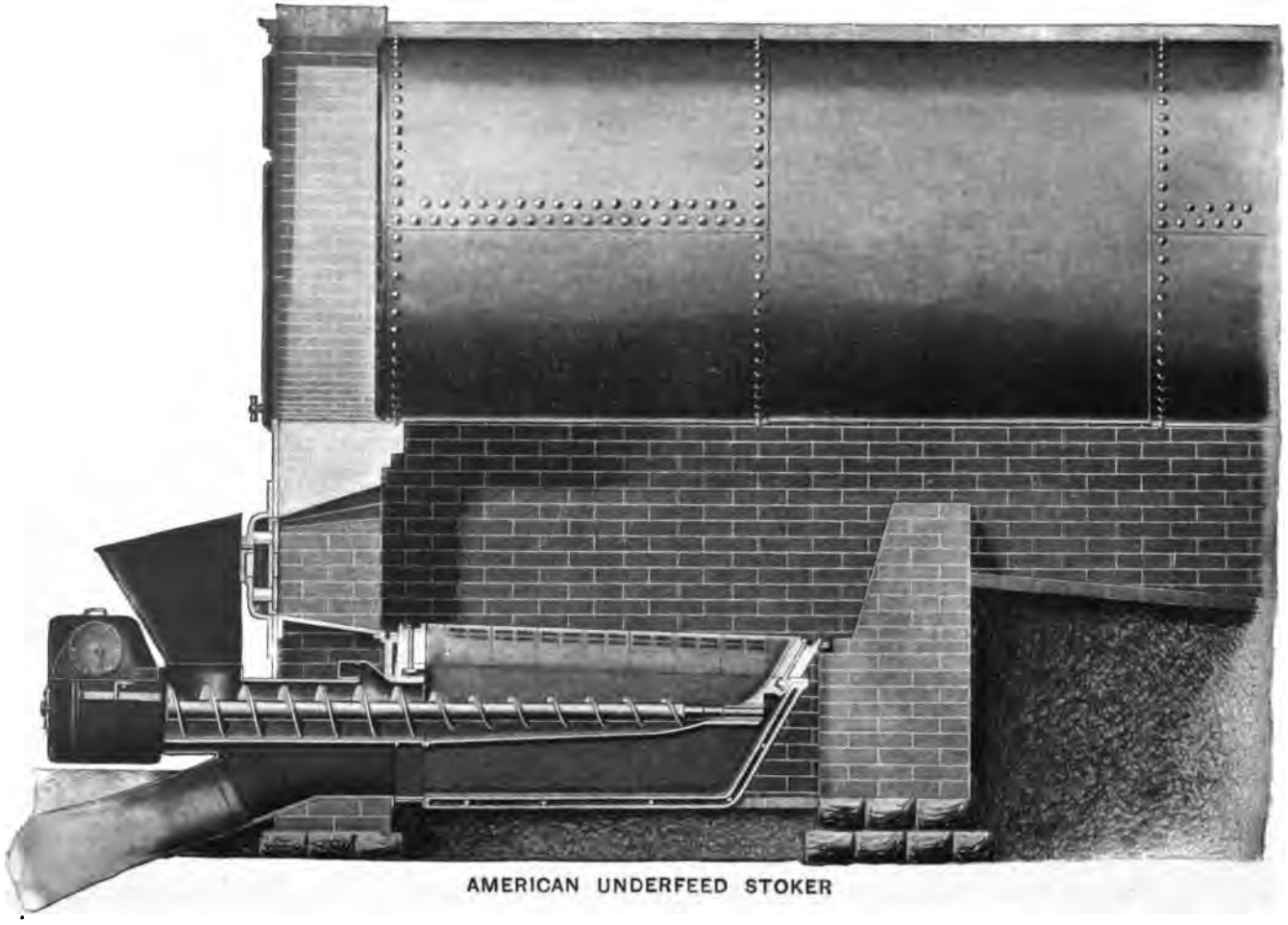


Fig. 4.

In the furnace made by the Hawley Down-Draft Boiler Co., the grates are formed of a series of water tubes, opening at the ends into steel drums, shown in Figs. 4 and 5, which are connected with the boiler. Fig. 4 shows this furnace attached to a



AMERICAN UNDERFEED STOKER

horizontal multitubular boiler. It may be applied to both *tubular* and *water tube* boilers with good results and is advantageous to boilers of insufficient heating surface and when inferior fuels are burned. It is claimed that this attachment insures complete combustion, small amount of ashes on account of the second grate, good water circulation and increased economy and capacity.

Fig. 5.

MECHANICAL STOKERS.

In order to save the labor of feeding furnaces by hand, mechanical stokers are used. With a good mechanical stoker, *one* man can attend several furnaces with little labor. There are several styles. The first was invented by James Watt. The coal after coking was pushed to the back of the furnace by levers.

One form consists of longitudinal bars connected by links forming an endless chain. The coal is delivered from a hopper at the front of the boiler to the grate. The moving of the grate from front to rear moves the coal into the furnace and drops the ash and clinkers at the back.

In the Roney stoker the fuel is dumped into a hopper at the front. The grate bars, which extend across the furnace, form a series of steps. The fuel slides down these steps while it is burning, until it reaches the dumping grate at the bottom. Each grate bar is hung on pivots at the ends. By means of a small steam engine the rocker bar (to which each grate bar is attached), tips the grate bars so that the upper surfaces incline downward toward the bridge, and then rights them. This motion causes the fuel to slide downward and then stop, because checked by the righting of the bars. The rate of feeding is controlled by the stroke of the "pusher," or by changing the number of strokes per minute of the pusher, or rocker bar. The ashes, clinker and the unburned

coal, if any, collect on the dumper grate, which is controlled by a lever.

If the rate of feed is too high, unburned coal will appear on the dumping grate. If the rate is too low, no unburned coal will be seen on the dumping grate, and the fire may be thin at the bottom, which will cause an excess of air to enter. The fireman is more likely to be uneconomical by using too little coal than too much, for he can see when unburned coal appears, but does not know the amount of waste caused by a thin fire.

All mechanical stokers in which the movable parts are inside the furnaces, are likely to get out of order because of the heat and dirt.

FUEL ECONOMIZERS.

Of the many attempts to get heat from the products of combustion after they leave the boiler and before they enter the chimney, Green's Economizer, Fig. 6, has been the most successful.

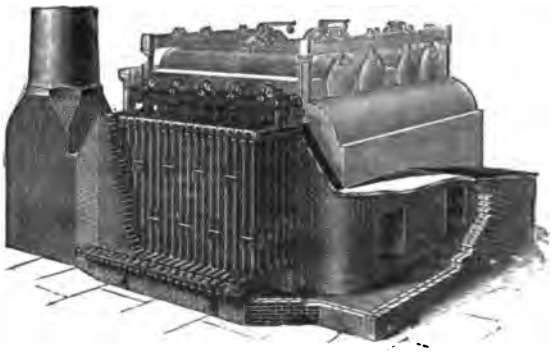


Fig. 6.

Most economizers have been unsatisfactory because of the accumulation of *soot* on the pipes. In Green's feed water heater, or economizer, the several sets of four-inch vertical tubes are kept clean by **scrapers**, one on each tube. By means of a small engine the scrapers move continuously up and down on the tubes. The feed water is pumped through several sets of tubes beginning with the most remote.

SPECIAL FURNACES.

Almost any furnace is adapted to burn anthracite coal, or semi-bituminous coal with less than 20 per cent. volatile matter.

If there is over 20 per cent. volatile matter in the coal, it should never be burned in a furnace that is directly under the boiler whose heating surfaces are exposed to radiation from coal on the grate. If the coal contains over 40 per cent. volatile matter the furnace should be surrounded with fire bricks and have a large combustion chamber.

Some variety of coal is generally used for fuel; but in some localities and industries wood, saw-dust, tan-bark, bagasse and straw are much cheaper or convenient. When these fuels are burned there should be plenty of room in the furnace and sufficient air supplied to the top of the fuel. Saw-dust and fine coal are sometimes burned by blowing them into the furnace by means of an air blast.

When oil is used it is thrown into the furnace by **sprayers** or **atomizers,** the oil burning in a long flame. The sprayer is formed of two concentric conical tubes. Compressed air or steam entering by the inner tube, draws oil through the outer tube and throws it into the furnace in the form of a fine spray. Steam is often used in place of compressed air on account of its convenience, but it is not as good as compressed air. The oil is usually stored in tanks, which are outside the fire room, and which should be lower than the sprayers.

Experiments show that one pound of petroleum is equal in heat units to about 1.8 pounds of coal. Petroleum has many advantages as a boiler fuel. It is clean, gives out uniform heat, and is economical. As there are no ashes to handle and no fires to clean, one man can tend several furnaces. The steady temperature and the freedom from admission of cold air by continual opening of doors increases both the economy and durability of the boiler. The fire may be started or stopped instantly, and the supply of air regulated accurately.

Petroleum has an unpleasant odor, is likely to explode, and has a comparatively high value. Economy from using petroleum or coal depends upon the locality, since prices vary with the difficulty of obtaining them.

NATURAL AND FORCED DRAFTS.

The draft in a chimney is caused by the difference in weight between the hot gases inside and the air outside. The force, or

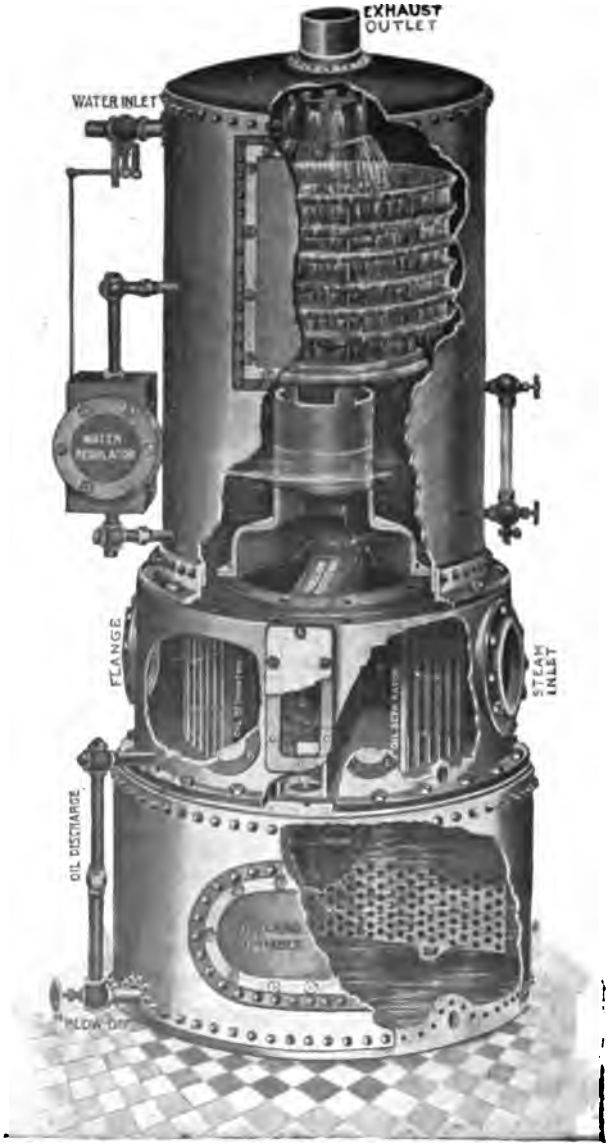
intensity of the draft is equal to the difference of these weights, and is measured by a draft gauge. One form of this instrument is a tube bent in the form of the letter U. The tube is partially filled with water, one leg being connected to the interior of the chimney and the other open to the external air. The difference of the water levels in the two legs indicates the difference of pressures and the amount of draft.

On account of the slight movement of the fluid and the error caused by the water being attracted to the dry tube, the Eames Differential Draft Gage is used. The form is shown by Fig. 7.

Fig. 7.

The fluid used is a special non-drying and non-evaporating oil of known specific gravity. The incline and diameter of the tube are so proportioned that the reading represents distilled water in hundredths of an inch. The indications are taken on one leg only. The instrument is capable of measuring two inches of water pressure.

The rate of combustion depends upon the height of the chimney. A chimney 20 to 25 feet high will cause draft sufficient to burn about eight pounds of coal per square foot of grate area per hour; if the height is increased, the rate of combustion is increased. If the chimney is about 100 feet high, the rate of combustion will be about 10 pounds per square foot of grate area per hour, and to burn 25 pounds, the chimney must be about 175 feet high. The height of the chimney varies with the fuel and the kind of boiler. For good bituminous, or anthracite coal, the chimney must be higher than for wood or poor bituminous coal, if the same rate of combustion is desired. If the boiler has small or winding passages the chimney must be high. As high chimneys are costly it has become common practice to build two or three small chimneys in place of a large one.



COOKSON FEED WATER HEATER.

It is often inconvenient and sometimes impossible to build high chimneys or several chimneys when a high rate of combustion is desired. In locomotive and marine work, and in large cities, a high chimney is undesirable. In order to obtain the high rate of combustion with the short stacks of locomotives and ships, a **forced draft** is used. By this means a high rate of combustion is obtained, — from 100 to 150 pounds of coal per square foot of grate area per hour. Forced draft greatly increases the power, but is likely to injure a boiler, and is uneconomical under some conditions.

There are three systems of forced draft. The induced draft used in locomotives, the closed stoke-hole and closed ash-pit used in marine boilers.

Induced Draft. In the first system, the exhaust steam is turned into the smoke stack. The force of the steam draws the products of combustion up the stack. Induced draft is also obtained in land boiler plants by placing a blower in the chimney. An economizer should be used so that the gases may be cooled before they reach the blower. The draft obtained on locomotives is equivalent to a column of water five to seven inches high.

Forced Draft. On steamships forced draft may be obtained either by making the stoke-hole air tight and then forcing air into it, or by increasing the pressure of air in the ash-pit.

By the first method, the stoke-hole is closed so that the only outlet for the air is through the furnaces. Air from without is forced into the stoke-hole until its pressure exceeds that of the atmosphere. This method gives good ventilation to the stoke-hole; but when the fire door is opened there is a great rush of air to the surface of the coal, the cold air chilling the furnace plates. With this form of forced draft the fireman must act quickly.

The closed ash-pit is used successfully both in the merchant service and on war-ships. Air is drawn from without by means of a blower, and is forced into the ash-pit, which has an air tight door. If the pressure is considerable, the fire door must also be air tight, or the flame may come out around the door. When the fire door is opened to admit coal, the draft must be shut off or flame will come out into the fire room. An automatic arrangement, which shuts off the draft when the door is opened, is often used.

The forced draft for marine boilers is usually equivalent to about two inches of water.

Blowers. For ventilation, heating and forced draft, blowers

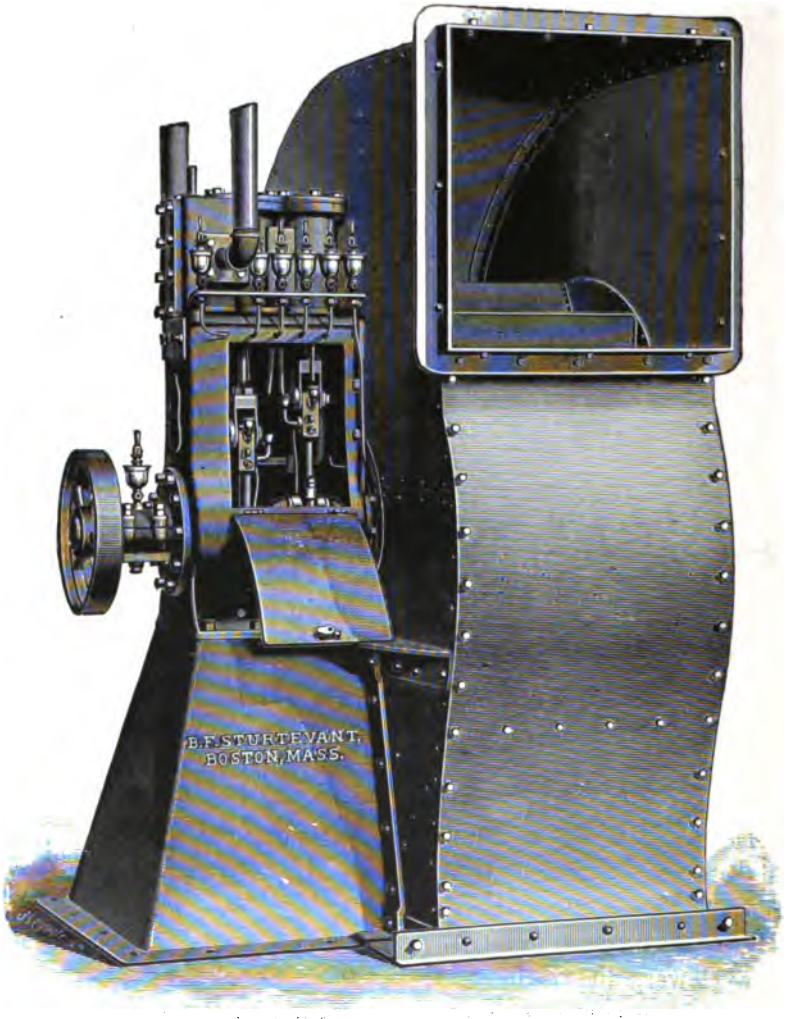


Fig. 8.

are generally used. In case induced forced draft is desired, a fan or exhauster is placed in the chimney to draw the gases through the passages. With forced draft, either with closed ash-pit or closed stoke-hole, a blower forces air into the ash-pit or stoke-hole.

Fig. 8 shows a Sturtevant blower with enclosed engines. A blower is a fan wheel encased in sheet iron, brick or wood having two inlets for air. An exhauster is constructed like a blower and has but one inlet. They are made to discharge vertically, horizontally, etc., and are constructed with steel plate fans and designed to handle air, dust, smoke, and steam with low resistances. For forced draft, they are capable of producing air pressure equal to four inches of water, and are very generally used with economizers, and in marine work.

VALVES.

The fittings used to regulate the flow of fluids in pipes are operated by moving a disk across the pipe, with or without rotation, or by turning it through an angle. Those operated by the first method are called valves, and those by the second, cocks.

The Globe Valve, shown in Fig. 9, gets its name from the shape of the case, which is separated into two parts by a partition with an opening through the horizontal part. The fluid enters at the right and passes up through the opening and out at the left. It is closed by screwing down the valve face to the seat by a handle. A stuffing box around the valve prevents leaking. In Fig. 9 the valve is round and is called the disc valve. The valve is guided to the seat by a smaller ring on the valve face. Other forms of globe valves have bevelled seats and the valve face is made of composition. Globe valves should be set so as to close against the flow; when set in the opposite way, if the valve should be detached from the stem it could not be opened, although the handle might move all right. It would also be necessary to shut off

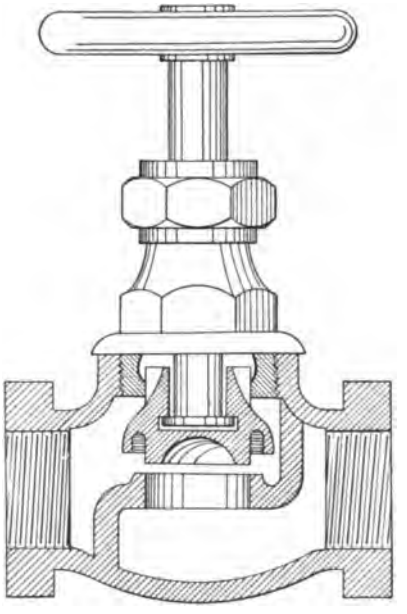


Fig. 9.

the fluid from the entire pipe to pack the valve. This valve offers considerable resistance to flow, since the fluid must make two turns at nearly right angles.

The Angle Valve is sometimes used to take the place of an elbow and stop valve. The inlet is at the bottom and the outlet at the side. The valve is guided to the seat by a rib or ring, the bearing surface being a ring of soft metal. The body of the valve is made of cast iron, the seats, stuffing box, etc., being brass or composition.

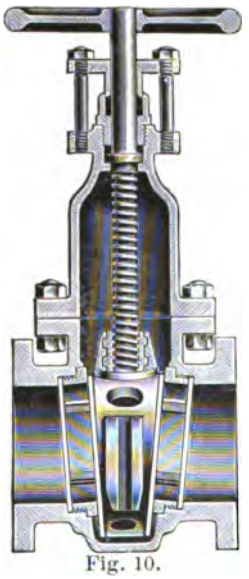


Fig. 10.

The Gate Valve, or straight way valve, gives a straight passage and offers very little resistance to flow when it is open.

There are two forms, one, the wedge shaped valve with wedge-shaped seats and the other having the valve seats parallel. Fig. 10 shows a Chapman valve with wedge-shaped seats. A collar holds the valve spindle at a fixed height. The valve is drawn up or lowered, to open or close, by turning the spindle. When the gate reaches the bottom of the pipe it moves laterally, sufficient to bring a

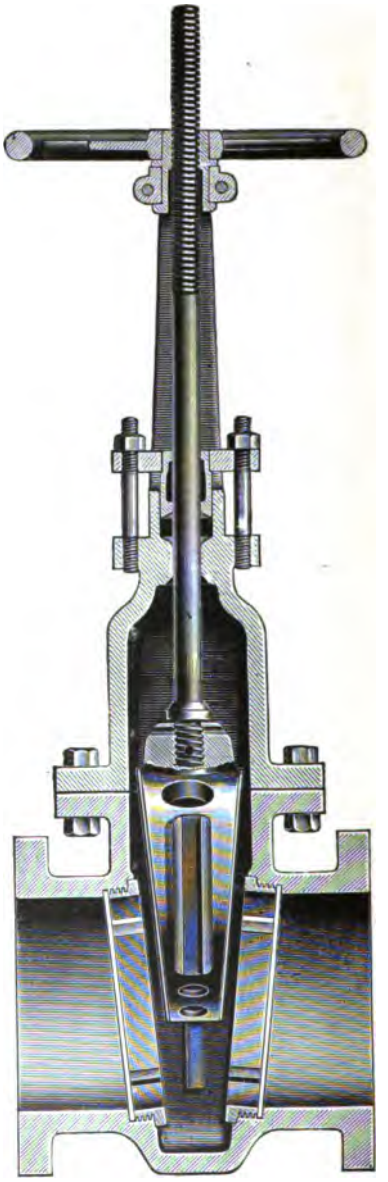


Fig. 11.

strong pressure against the seat. These valves are made with a **stem** which rises with the gate, as shown in Fig. 11, or with one which remains in the same position whether the gate is open or closed. The *former* is preferable as one can tell at a glance whether the gate is open or shut.

Check Valves. When it is necessary that the flow should always take place in the same direction, as in the feed pipe of a boiler, check valves are used. There are several forms shown in Fig. 12, one of which has a similar pattern to a globe valve, with a ball or flat valve, the seat being parallel to the direction of flow. The valve is held in place by its own weight and by the pressure of the fluid, in case of a reverse flow. In the swinging check

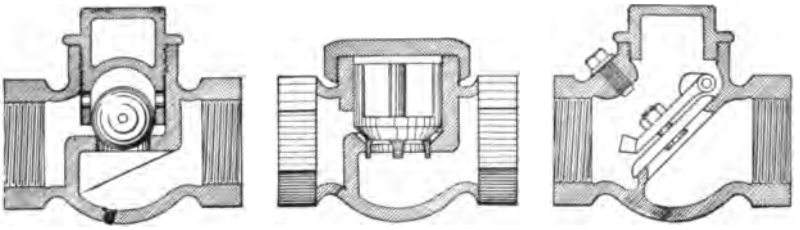


Fig. 12.

valve, the seat is at an angle of about 45 degrees to the direction of flow. It is fitted somewhat loosely where it is fastened to the swinging arm, so that it may properly seat itself. This form is usually preferred, as it offers less resistance to flow and there is less chance for impurities to lodge on the valve seat. When a check valve is used in the boiler feed pipe, there should be a *stop valve* between it and the boiler, which can be shut in case the check valve gets out of order.

SAFETY VALVES.

Safety valves are used for reducing the pressure when it exceeds a certain limit and also to give warning of high pressure. They are constructed in several forms, but in every case they consist of a valve opening outward and held in place by a weight or spring.

A simple form is shown in Fig. 13. It consists of weights suspended from the valve and hanging inside the upper part of

the boiler and has a cup-shaped seat. This form is advantageous because the weight cannot be altered without opening the boiler. The *lift*, or the distance the valve rises, is rarely more than $\frac{1}{8}$ of an inch. The area for the escape of steam should be sufficient to discharge all steam that the boiler can make. The area can be computed by an empirical formula proposed by Rankine.

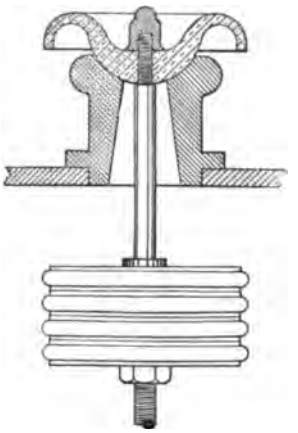


Fig. 13.

$$A = \frac{70\ W}{p},$$

in which A is the effective area of discharge, W the weight of steam delivered per second, and p the absolute pressure in pounds per square inch.

The amount of steam depends on the area of grate, rate of combustion and the evaporation per pound of coal. The first condition is known and the other two may be estimated from the prevailing conditions and from the type.

Suppose the boiler makes $1\frac{1}{2}$ pounds of steam per second, with 95 pounds absolute pressure. Then the area will be,

$$A = \frac{70 \times 1\frac{1}{2}}{95} = 1.1+ \text{ square inches.}$$

Suppose the lift to be $\frac{1}{10}$ of an inch, the circumference will be $1.1+ \div .1 = 11$ inches and the diameter will be about $3\frac{1}{2}$ inches.

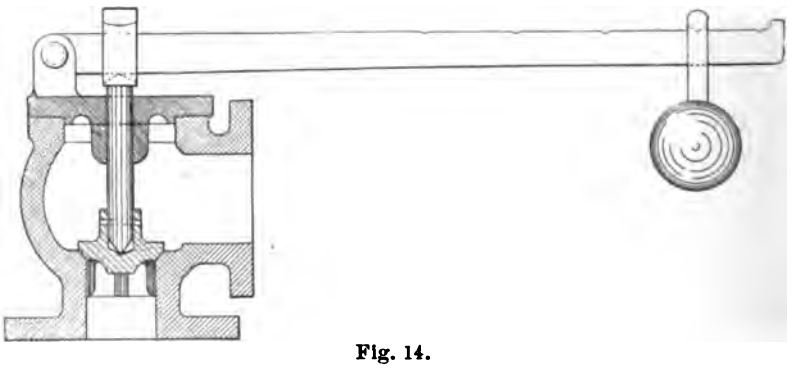


Fig. 14.

Another rule is to make the area of the valve seat $\frac{1}{3}$ of a square inch for every square foot of grate surface.

If more than one method is used in the calculation, the largest result should be used.

Another form much used is the lever safety valve, shown in Fig. 14. The valve is held in place by a weight on the end of a *lever*. The force required to lift the valve is regulated by sliding the weight on the lever. The body of the valve is cast iron and has an opening at one side to allow the steam to escape. The valve and valve seat are of brass. When the valve becomes leaky, it should be reground and made tight, but some carelessly try to stop the leak by hanging on extra weight, or by wedging it down. For this reason it is well to have the lever cut off at the proper length and to use a short lever and heavy weight.

The method of calculation for the lever safety valve is the same as that for any lever. The safety valve is a lever of the third class because the power (the steam pressure) is between the fulcrum and the weight. In any lever the *weight*, multiplied by the *weight arm*, equals the **power** multiplied by the **power arm.** In the lever safety valve the power is the steam pressure, and the power arm is the distance of the center of the valve from the fulcrum. There are three weights, since the ball, the lever and the valve and spindle all act downward; their respective arms are the length of the lever, the distance of the center of gravity of the lever from the fulcrum, and the distance between the center of the valve and the fulcrum.

Thus if we know seven of the following eight conditions, we can find the eighth.

The weight of the ball at the end of the lever.

The length of the lever.

The weight of the lever.

The distance of the center of gravity of the lever from the fulcrum.

The weight of the valve and spindle.

The distance of the center of the valve from the fulcrum.

The diameter of the valve.

The distance of the valve from the fulcrum.

Suppose we have a lever safety valve as shown below. We wish to know at what pressure per square inch it will blow. We know that the product of the total steam pressure on the valve,

multiplied by the distance of the valve from the fulcrum, equals the sum of the three weights each multiplied by their respective lever arms, or

$$\begin{aligned} 125 \times 48 &= 6000 \\ 50 \times 20 &= 1000 \\ 18 \times 4\tfrac{1}{2} &= \quad 81 \\ &\hline \\ &\quad 7081. \end{aligned}$$

$$\text{Area of valve} = \frac{\pi d^2}{4} = \frac{3.1416 \times 25}{4} = 19.635.$$

Let p = steam pressure per square inch; then $19.635\ p$ = the total power. From the law previously stated

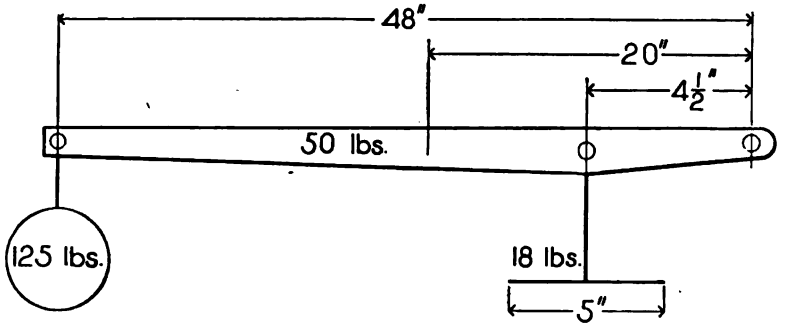
$$19.635\ p \times 4\tfrac{1}{2} = 7081.$$

$$88.357\ p = 7081$$

$$p = \frac{7081}{88.357}$$

$$= 80 \text{ pounds.}$$

Then we may say that the safety valve will blow at 80 pounds per square inch.



EXAMPLE FOR PRACTICE.

A lever safety valve has the following dimensions: Valve $4\frac{1}{2}$ inches in diameter, weight of valve and spindle 14 pounds, distance of valve from fulcrum $4\frac{1}{2}$ inches, weight of lever 44 pounds, center of gravity of lever 18 inches from fulcrum, lever 42 inches long. What should be the weight of the ball for the valve to blow off at 60 pounds pressure?

Ans. 82 pounds nearly.

If the pressure at which the valve will blow is known, and the size of the valve is unknown, it may be found by the application of the same principle.

The center of gravity of the lever can be calculated, but a much easier method is to balance it over a knife edge and note the position of the knife edge.

In the above calculations, the theoretical pressure only is calculated. The friction of the various joints may alter the blowing pressure somewhat. A safety valve should be set by trial to blow off at the required pressure. In setting the safety valve a pressure gage should be used that has been compared with a standard gage or mercury column.

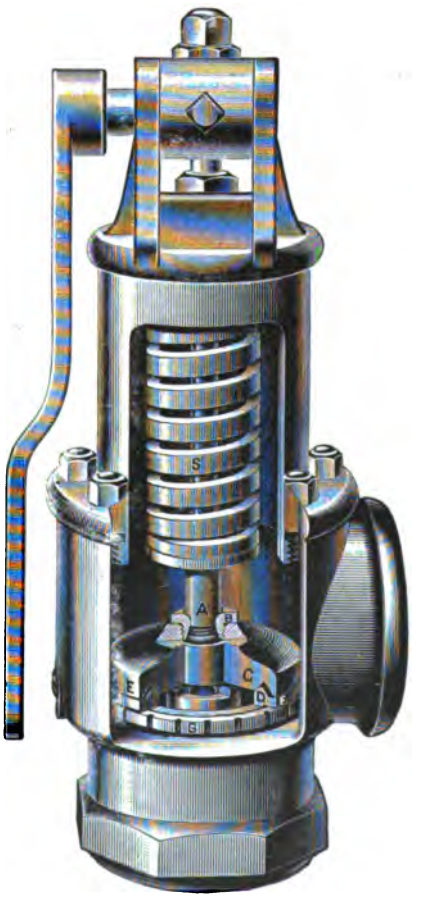


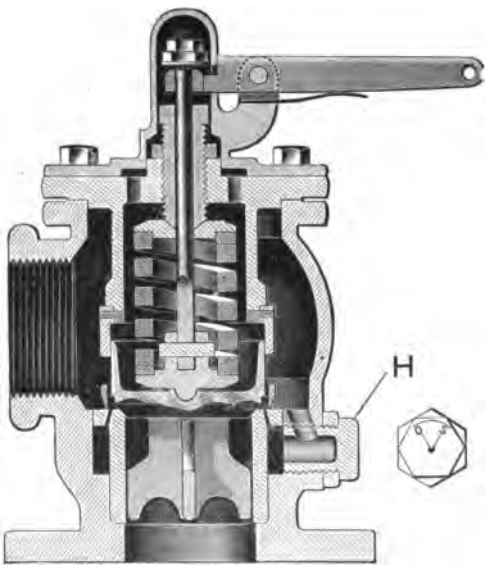
Fig. 15.

The common lever safety valve has several *defects*. It does not close promptly when pressure is reduced; it is likely to leak after it is closed and it can be easily overloaded or wedged on its seat. The safety valve which is automatic, certain in action, prompt in opening and closing at the required pressure, and which can be relied upon to relieve the boiler under all circumstances, is the one desired. The spring, or "**Pop**" **safety valves** fulfill the above requirements better than those of the lever type. The valves open when the steam pressure is sufficient to overcome the tension of the spring. Fig. 15 shows a Crosby Pop Safety Valve for stationary service. The valve C, the main valve, is connected by the flanged nut B to the central spindle A, and is held down

on its seat by the pressure of the spring S. The valve C is provided with wing guides and an annular lip E. The guides fit smoothly into the seating D, upon which the valve rests. The seats of the valve have an angle of 45 degrees. The under face of the lip E together with the seating forms a small chamber through which all the steam must pass to the open air. A number of small holes drilled vertically through the flange F, connect with the chamber and allow part of the steam to escape. The action of the valve is regulated by the screw ring G, which allows more or less steam to escape through the holes in the flange F. Raising the screw diminishes, and lowering it increases, the area of the holes. If the loss of steam is too great when the valve blows, turn the screw ring down.

The safety valve should be connected **directly** to the boiler without any pipe or elbow. It should be tried every day by means of the lever.

The valve shown in Fig. 16 for stationary boilers, is made by the Ashton Valve Co. The general principles are those of all pop safety valves. The valve seat is made of composition, or nickle, and with a bevel of 45 degrees, as is the United States Government standard. The pop chamber is surrounded by a knife-edge lip, which wears down in proportion with the seat, thus keeping the outlet of the same relative proportions, giving a constant amount of pop.



Sectional View. Fig. 16.

The amount of pop, that is, the difference of pressure between the opening and the closing of the valve, is regulated from the outside by means of the screw plug pop regulator shown at H in

Fig. 16. If more pop is desired, turn the regulator so that S will be more nearly perpendicular. To lessen pop, make O more nearly perpendicular. The springs are made of Jessop's best steel.

The inlet and outlet are both on the same casting, so that the valve may be taken apart to be cleaned or repaired without disturbing the boiler connection. It has a lock-up attachment, so that the regulating parts can not be tampered with, either by accident or design. The spring is encased, thus protecting it from the steam.

The Star Marine Pop Safety Valve is shown in Fig. 17. It

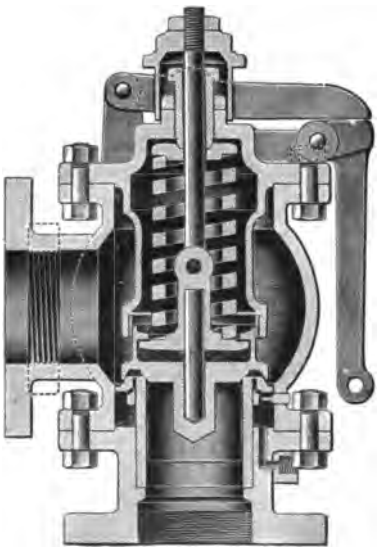


Fig. 17.

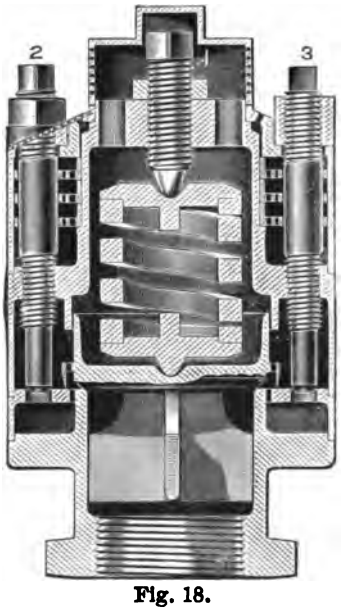


Fig. 18.

has a bevel seat and is provided with a cam lever by which it may be raised from its seat when there is no steam pressure. The outlet of the valve can if desirable, be piped to the supply tank or any other point.

The safety valve for *locomotive boilers* must be made of heavy material to stand the severe usage. They should be so constructed that they will not cock or tilt. The Ashton valve shown in Fig. 18 is constructed so that the amount of pop may be regulated by merely turning the two posts marked 2 and 3 to the right or left. The noise of the steam escaping from the ordinary safety

valve is disagreeable, and in some states the law requires the use of the muffler safety valve. The Ashton valve shown in Fig. 18 has a top muffler.

REDUCING VALVES.

Sometimes steam is desired at a lower pressure than that of the boiler. For instance, a small low pressure engine may be run by steam taken from the same boiler that supplies a higher pressure engine. This reduction is accomplished by throttling the steam by means of reducing valves. They are arranged to be operated automatically so that the pressure may be reduced and a constant pressure in the steam pipes main-

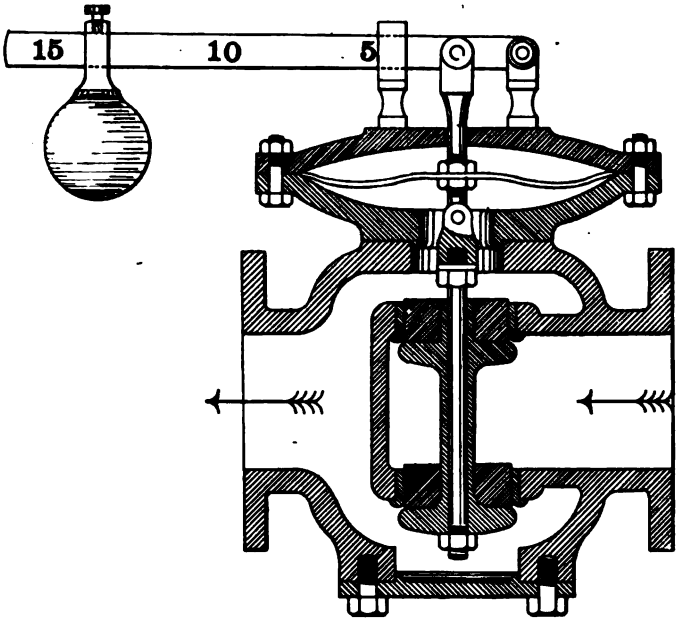


Fig. 19.

tained. There are several forms in general use. Let us consider the action of a few. In the Holt valve, Fig. 19, the low pressure steam acts on the lower side of the diaphragm and the weight, which may be set so as to cause the desired pressure, acts on the other. The movement of this diaphragm causes a balanced valve to move to or from its seat. The valve opens until the steam pressure equals the weight above. The pressure in the

main steam pipe does not effect the movement of the valve. It depends only upon the pressure on the two sides of the diaphragm.

Another form, the Mason, is shown in Fig. 20. A spring, which may have its tension altered by a key, takes the place of the lever and weight, in the Holt valve. When the pressure in the low pressure system has risen to the required point, which is determined by the spring, the valve closes and no more steam is admitted until the pressure falls sufficiently to open the valve again.

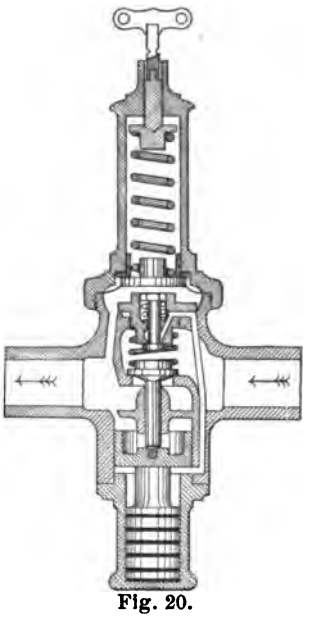


Fig. 20.

In another form, a piston acted on by the low pressure steam regulates the opening of a balanced valve, and this maintains a constant steam pressure.

In the Foster reducing valve, the valve is held open by the spring and levers, until the steam pressure at exit presses on the diaphragm sufficiently to close the valve. The valve is held open so as to admit just the proper amount of steam to maintain the required pressure.

When a reducing valve is used, a stop valve should be put in to prevent flow when steam is not in use. It also regulates the supply until it exceeds the leakage of the reducing valve.

BLOW-OUT APPARATUS.

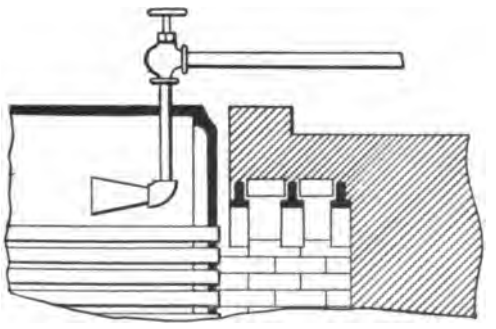


Fig. 21.

Water for feeding boilers, if taken from rivers or ponds, is likely to contain some vegetable matter. As this matter usually floats on the surface of the water in the boiler, it may be discharged through a **surface blow-out.** This apparatus, shown in Fig. 21, consists of a flattened cone of sheet

metal extending nearly across the boiler and turned so that the currents will deposit solid matter in its mouth. The valve is opened frequently to blow out these vegetable impurities.

There are many solid impurities that will settle to the *bottom* of a boiler if the water is allowed to remain quiet for a short time. By means of a mud drum, or a **bottom blow-out** apparatus, these solid particles are removed. In many boilers, the blow-out cocks are kept open every morning long enough to reduce the level of water in the boiler a few inches. This blow-out apparatus is situated near the bottom of the boiler, in order that it may be completely drained. *Valves* are more easily opened than *taps*, but are not as trustworthy. It is almost impossible to completely close a valve if a small piece of scale, a small chip, or an accumulation of sediment lodges on the seat. If the valve is not closed tightly, water will leak out; the boiler may be drained, and then become overheated. One is not deceived as to whether a tap is closed or open. For these reasons it is better adapted for blow-out apparatus than a valve. A tap is likely to stick fast because of corrosion, or unequal expansion, but this difficulty is slight if it is opened *frequently* and *gradually*. The plug and casing should not be made of the same metal, because they will stick if they remain in contact for any length of time.

WATER GAGES.

It is of great importance that the level of the water in the boiler can be easily ascertained at any time. If the water is too **low** there is danger of **overheating** the furnace plates; if it is too **high** there is likely to be **priming**.

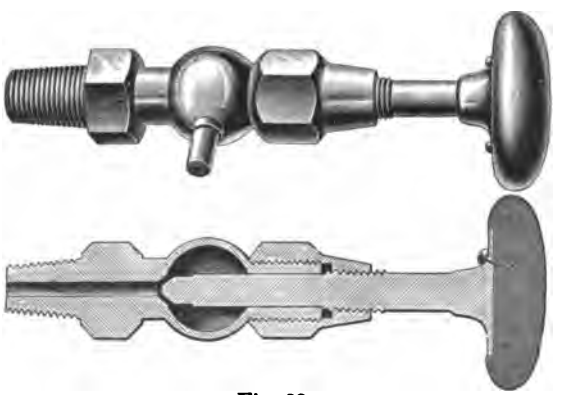


Fig. 22.

The water level is usually indicated by gage cocks, water gage glasses and floats.

Gage cocks, or **Try cocks**, are very generally used. They are of different forms, varying with the makers. Fig. 22 shows a form for locomotives. It is so constructed that it can be separated for the purpose of repacking the working parts without detachment from the boiler. The one shown in Fig. 23, used largely on stationary boilers, is made both with and without packing.

Fig. 23.

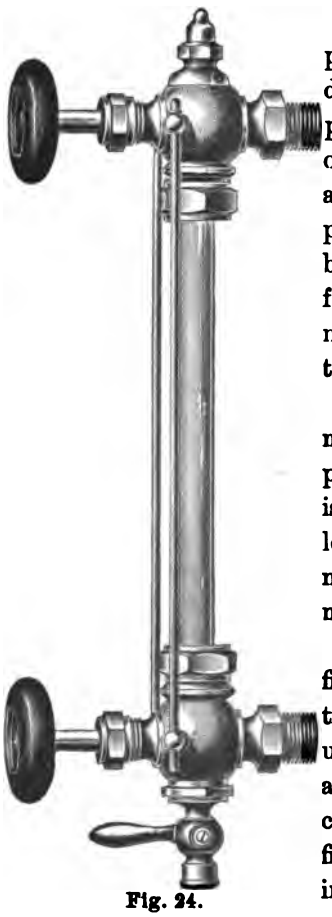


Fig. 24.

There are usually three cocks, one placed at the highest position that it is desired to have the water level, one placed at the lowest position and the other halfway between. For greater accuracy more cocks may be used. The position of the water level is determined by trying the cocks. If steam escapes from the highest one, and water from the next, the water level is evidently between the two.

In flat ended, multitubular, flue and marine boilers the cocks are usually placed on the *front* end. If the boiler is encased in brickwork above the water level, the water level is found by connecting the cocks with the boiler by means of small pipes.

Gage Glasses. In order that the fireman may know the water level *without* trying the cocks, a water gage glass is used. It consists of a strong glass tube about one foot in length having the ends connected to the boiler by suitable fittings. As both ends of the tube are in communication with the boiler, the water level in the glass will be the same as that in the boiler, and is **always in sight.** Fig. 24 shows

a good form of gage glass. The glass is protected by rods which are parallel to it. As the glass frequently needs cleaning, repacking, or renewing, cocks are provided for shutting off communication with the boiler. A drain cock also is placed at the lower end to empty the glass when the attendant wishes to ascertain whether the glass is working properly or not. The drain cock is often provided with a drain pipe. The steam and water passages should be at least one half-inch in internal diameter.

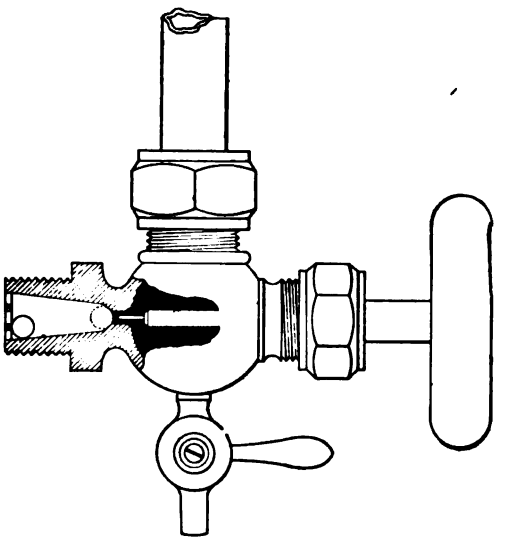


Fig. 25.

The glass is likely to break, because of accident or unequal temperature. To prevent serious injury to the fireman and loss of water as a result of the breaking of the gage glass, *automatic valves* are placed in the passages. In Fig. 25 the ball valve is shown in detail. If the glass breaks, the pressure of the steam drives the ball outward, filling the conical passage. When a new glass is put in, the balls are forced back by slowly screwing in the stems. This, like other safety devices, is very likely not to work when it should.

In boilers where the steam space is small, as in locomotives, the allowable variation of water level is slight, but the greater care with which the glass is watched makes up for the small margin of safety. If dirty water is used, or if the water foams,

the level in the glass will be unsteady and unreliable, since dirt clogs the passages, unless they are large, and the foaming causes a fluctuation of the water level. A small pipe connecting with the steam space where no ebullition occurs will insure a steadier water level. If the steam and water connections are long, they should be made large.

Two water gages should be used on every boiler, one as a check on the other, and also for use if one breaks. They should be placed at the furnace end of the boiler in order that they may be easily seen.

There are numerous arrangements for indicating the water level by means of floats. The most common is the counterbalance and pulley. Sometimes a whistle is attached to the float which gives warning when the water level becomes dangerously low. In using a float, the wire should never be made of iron, as it is liable to corrode, nor should it be large as it is more difficult to pack and more likely to be bent permanently. When carefully made and attended, floats are reliable water indicators. There should always be two, one at each end of the boiler.

STEAM AND VACUUM GAGES.

The steam pressure in the boiler is measured in **pounds per square inch.** When we say the boiler is working or steaming at 80 pounds pressure, we mean that the gage pressure is 80 pounds, that is, the pressure in the boiler is 80 pounds above **atmospheric pressure.** It could be measured by a water or mercury column, but as they would need to be very high to measure the pressures used at the present day, they are not practicable. A spring pressure gage is used instead.

Although not absolutely necessary for safety, it is very desirable that every boiler have a reliable steam gage to indicate the steam pressure.

The dial gage, now used almost universally, was invented by M. Bourdon. It is designed with reference to the principle that a flattened, curved tube closed at one end, tends to become straight when subjected to internal pressure.

The tube, which is usually oval in section, is bent into the arc of a circle as shown in Fig. 26. One end is *fixed* and is in

communication with the boiler. The other is *closed* and free to move. By means of levers, a curved rack and a pinion, the motion of the free end is *multiplied* and *indicated* by a needle which is attached to the pinion. The needle moves over a dial which is graduated to agree with a mercury column, or with a standard gage. The back-lash of the levers is taken up by a hair spring. Fig. 27 shows the interior and face of a Bourdon steam gage manufactured by the American Steam Gauge Co., and is called the Lane Gage.

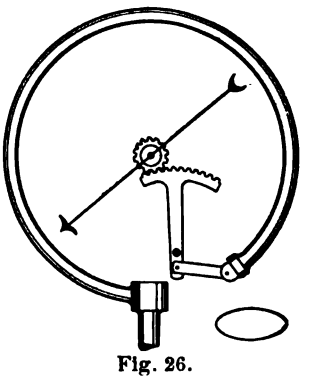


Fig. 26.

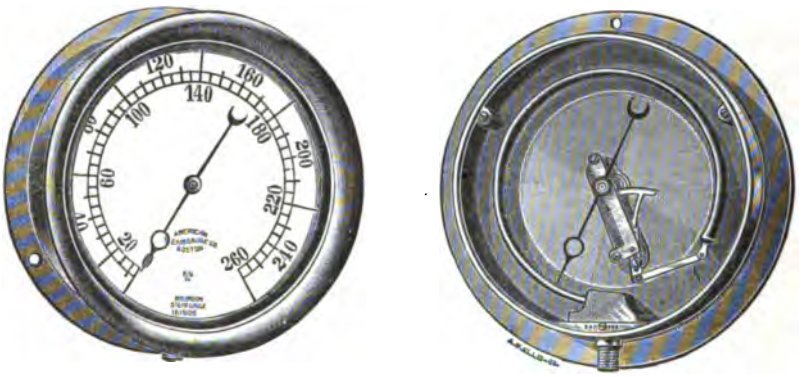


Fig. 27.

Fig. 28 shows the exterior and interior of a steam gage with a light tube for low pressures; the face is graduated corresponding to the mercury column. The only difference between this gage and the **vacuum gage** is that in the latter, the curved tube is turned in the opposite direction so that the needle will move clockwise with a *decrease* of pressure.

On account of the jarring, the gage for locomotives must be very strong. To prevent excessive vibration of the needle, two short stiffer springs are used as shown in Fig. 29.

Sometimes two pressure gages are fitted to a boiler; one indictating the working pressure and the other graduated to about twice the working pressure. The latter is useful in testing the boiler under water pressure and also serves as a check on the other.

The pipe which connects the pressure gage to the boiler should have

Fig. 28.

bends in it near the gage. These bends, or better a coil pipe as

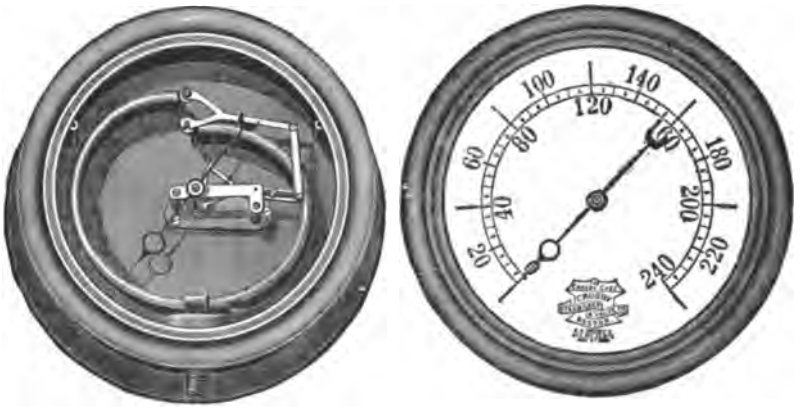


Fig. 29.

shown in Fig. 30, are filled with water which transmits pressure

Fig. 30.

and keeps the spring at a nearly constant low temperature. Gages

should be placed where the water in the coiled pipe will not freeze; also, the gage should not be exposed to strong heat. In order that the gage may be removed from the boiler, for examination, repairs, or calibration, when the boiler is under pressure, the connection should be provided with stop cocks.

In a battery of boilers, **each** should have its pressure gage. It should be connected **directly** to the boiler, **not** to the steam pipe.

FUSIBLE PLUGS.

Fusible plugs are frequently inserted in the plates of boilers as a safeguard against collapse of the furnace crown. They consist of a core of an alloy of tin, lead and bismuth, and a covering of brass or cast iron. While the alloy is kept at a comparatively low temperature by the water on one side, the fire on the other will not melt it. But when the water level becomes low enough to leave the plug uncovered, the core of alloy, having a low melting point, is expected to fuse, thus relieving the pressure in the boiler and extinguishing the fire. Fusible plugs are unreliable; blowing out when there is no apparent cause, and sometimes

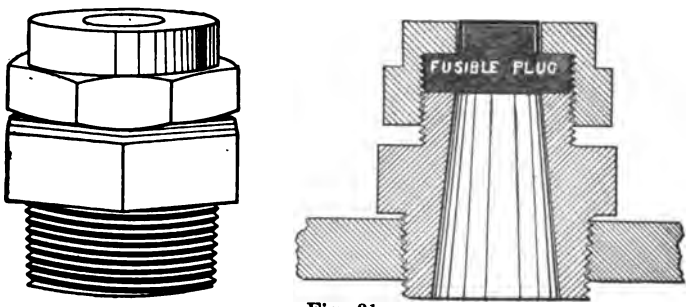


Fig. 31.

remaining intact when the plates become overheated. If a coating of hard scale is allowed to form over the fusible plug, it may stand considerable pressure even after the alloy has become melted. The nature of the fusible alloy may change if it is exposed to heat for a considerable time.

Fusible plugs should be cleaned from soot and dirt and the surface scraped frequently. They should be renewed every three or four months that no change may take place in the nature of

the alloy. The fusible plug should be made of such shape that when screwed into the crown-sheet it projects one and a half or two inches *above* the plates, so that when the alloy melts there will still be sufficient depth of water over the exposed plates to prevent injury from heat. The core is often made annular with a copper center. This gives a very large opening. Sometimes the core is covered with a thin copper cap, shown in Fig. 31, which protects the alloy from contact with the water, thus preventing

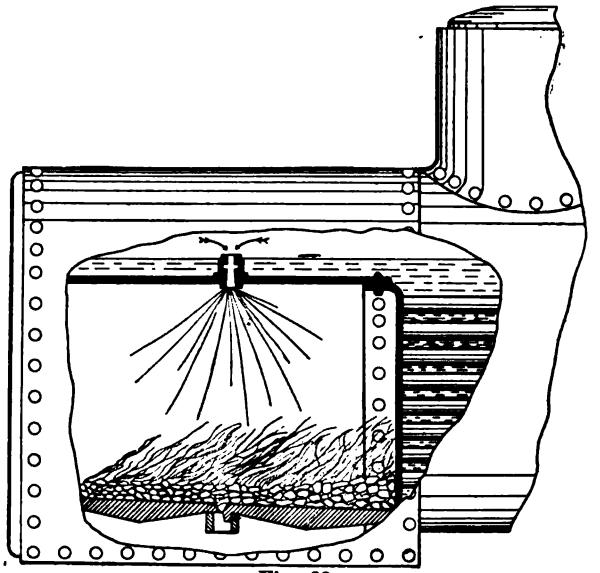


Fig. 32.

chemical change and the formation of scale. Fig. 32 represents a fusible plug in the crown sheet of a fire-box boiler.

Fusible plugs are placed in the plates which are directly exposed to great heat. The following are some of the places where a fusible plug is used:

In the crown sheet of a locomotive fire box.

In the lower tube sheet, or a little above the crown sheet, in one of the tubes of a vertical boiler.

In the back head of a cylindrical tubular boiler, about three inches above the top row of tubes.

In the lower portion of the upper drum of a water tube boiler.

It does not necessarily follow that a hole one half or three fourths inch in diameter will liberate steam fast enough to prevent excess of pressure. If a small quantity of steam is introduced into the fire box, the fire may brighten and the heat of combustion be greatly increased, because the steam will have an effect similar to that of forced draft. If, however, the quantity of escaping steam and water is considerable, combustion will be retarded and pressure relieved.

STEAM SEPARATORS.

The water surface of a boiler may not be sufficient for the *free disengagement* of steam, and so cause *wet steam.* Some water-tube boilers are defective for this reason. If steam is carried for some distance in a pipe, some of it will be condensed, and the water accumulating in a joint or sag in the pipe, is carried over to the engine when the pressure in the pipe increases slightly. This is likely to cause *injury* to the engine, besides being *uneconomical.* The amount of water thus formed in the pipe is greatly reduced by **superheating** the steam. If it is undesirable or inconvenient to superheat the steam, there are several ways to remove the water of condensation. One method is to let the steam come to rest in a drum of considerable size; the water is drained from the drum and the steam flows from the top by a pipe. If the drum is small the steam may not come to rest and the water will not separate from the steam.

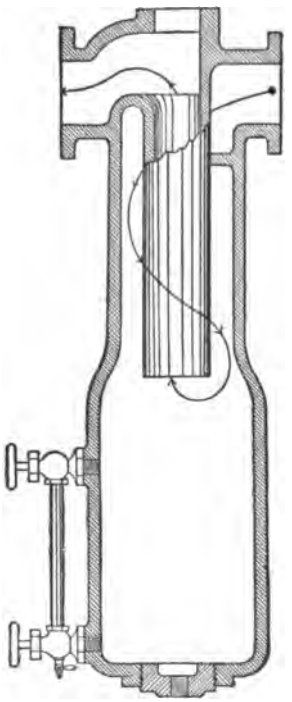
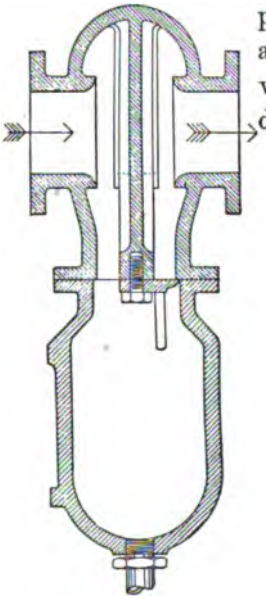


Fig. 33.

Another method is the use of a steam **separator.** There are several forms which are designed from the principle that if the direction of a current is suddenly changed, or if it goes downward then upward the water will separate from the steam and fall to the bottom.

In the Stratton separator, Fig. 33, the steam enters at one side of a cylinder, flows downward and then upward through a pipe in the middle. Dry steam escapes from a pipe near the top on the opposite side from which it enters. The separated water is drained at the bottom.



The Cochrane steam separator, shown in section in Fig. 34, is of the baffle plate type. The branches for the entrance and exit of the steam project from each side of the spherical head. Another branch from the bottom provides for connection with the well. The baffle plate, which is cast as a part of the head, is ribbed, or corrugated, and has ports at each side for the passage of steam. The area of the ports is large to prevent loss by friction. A small pipe is inserted in the lower plate of the outlet side of the baffle plate to drain any condensation in the outlet chamber. Steam entering at the left hand opening strikes the baffle plate and passes to the outlet chamber by means of the two side passages, as shown in the plan of Fig. 34.

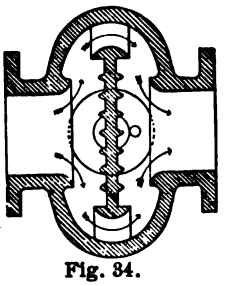


Fig. 34.

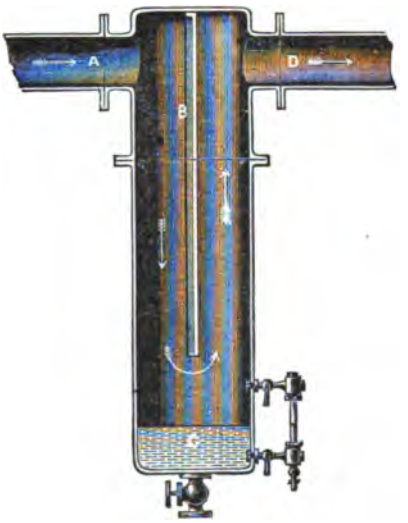


Fig. 35.

A form of separator which is fitted to the main steam pipe near the engine is shown in Fig. 35. Steam enters at A and

strikes the dash plate B; any water coming with the steam is separated and falls to the bottom. The steam takes the direction indicated by the arrows and flows out at D. This separator is fitted with a gage glass which is similar to a boiler gage glass.

STEAM TRAPS.

Steam traps are used to drain water from steam pipes. Traps usually consist of a vessel, or receptacle, into which water drains. The exit is a valve controlled by a float, which arrangement allows water but not steam, to pass.

In the **float** trap shown in Fig. 36 the float rises and falls with the change in the level of the water. When the water

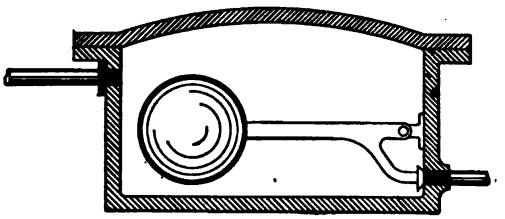


Fig. 36.

level rises above a certain point, the float opens the discharge valve. The trap shown in Fig. 37 is similar, the float being

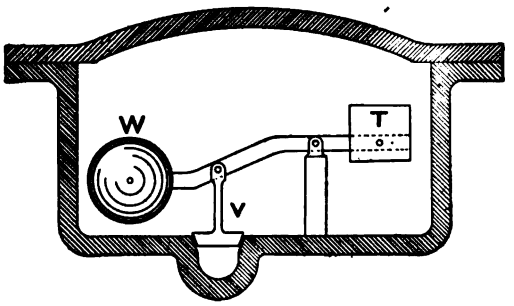


Fig. 37.

replaced by a weight W which is nearly counterbalanced by the weight T. The rising of W opens the valve V.

There are other forms called **bucket** traps. In the one shown in Fig. 38 the water enters at W. While there is a little water around the bucket F, it floats and the valve V is closed;

but when the water rises high enough to flow over the edge, the weight of water in the bucket causes it to sink and opens the valve V. Water is forced up the passage M and out the pipe N

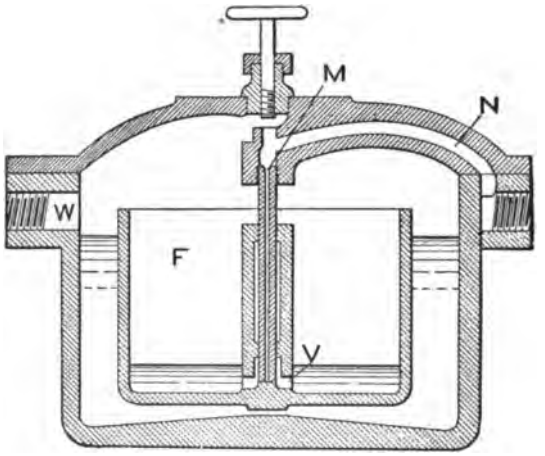


Fig. 38.

by the pressure of the steam on the surface of the water which surrounds the bucket.

Another form of trap called the differential steam trap depends upon a head of water acting on a flexible diaphragm. Water enters at either top or bottom by the pipes E, Fig. 39. When the water level rises it fills the chamber G and the pipe N. This causes a pressure on the under side of the diaphragm greater than that caused by the spring H, which acts on the upper side of the diaphragm and tends to open the valve. When the water rises and fills the chamber J so as to flow down the pipe M, the pressure on the upper side of the diaphragm will be greater than the pressure on the lower side because the head of water in M is greater than

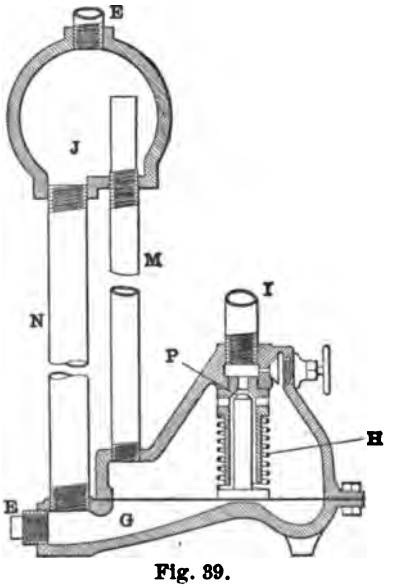


Fig. 39.

that in N. Then the spring will open the valve P and water will be discharged from the pipe I. When the head in M falls, the pressure on the under side of the diaphragm becomes greater and the valve closes.

Return Traps. Traps that are used for returning water of condensation to the boiler are called return traps. There is a variety of form, but the principle of action is similar and is shown in Fig. 40. B represents the boiler and T the trap, which is

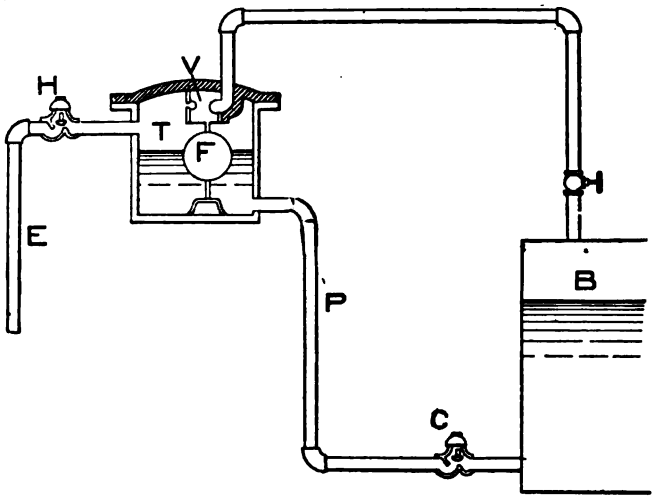


Fig. 40.

placed a few feet above the boiler. The trap is supplied with steam from the boiler. It is also connected with the boiler by the pipe P in which is a check valve at C. Water of condensation enters the trap through the pipe E, in which is a check valve H, until it reaches a depth sufficient to raise the float F, which opens the balanced steam valve V, called an equalizing valve. Steam from the boiler then enters the trap and equalizes the pressure. Since the pressures are equal, water in the trap, because of its height above the water level of the boiler, will flow to the boiler until the level in the pipe P is nearly the same as the water level of the boiler. As the water level in the trap falls, the float F drops and the equalizing valve is closed. In other forms buckets are used instead of floats.

Steam Domes. Steam domes or chambers are attached to

boilers for the purpose of obtaining dry steam and as a remedy for priming. They are supposed to be reservoirs to store a supply of steam, that may meet any sudden demand. The cutting away of a large piece of plate weakens the boiler considerably. Domes, if used, should be as small as convenient and the boiler plates around the hole should be strengthened as for manholes.

SUPPORTS.

There are two common methods of supporting boilers; *1.* By means of brackets; *2.* By suspending from wrought iron beams.

If the boiler is about 15 feet long, it is customary to use two

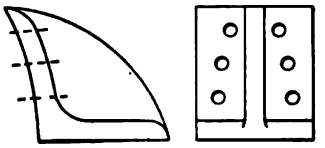


Fig. 41.

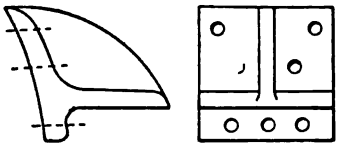


Fig. 42.

brackets on each side. If more than 15 feet, three on each side are used. The front brackets rest on the brickwork, but the others rest on small iron rollers to allow for expansion. Brackets are so arranged that the plane of support will be a little above the middle. There are several forms of brackets. The form shown

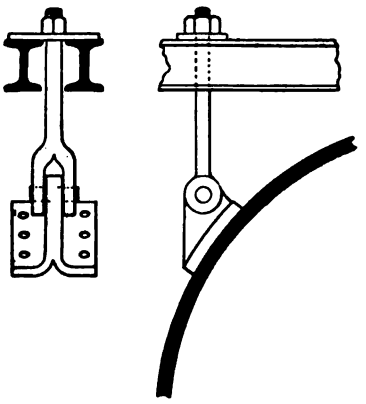


Fig. 43.

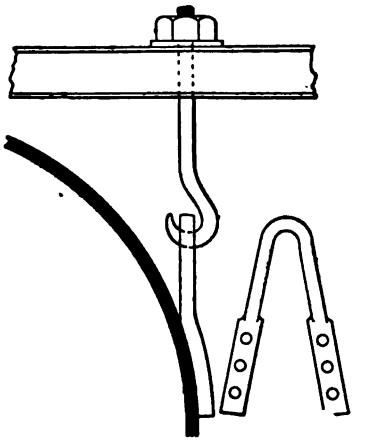


Fig. 44.

in Fig. 41 is usually made of cast iron and is provided with rivets above the flange of the bracket. It is better to have the rivets both above and below the flange as shown in Fig. 42.

Fig. 43 shows the method of suspending from beams. The lug, made of wrought iron, is riveted to the plates of the boiler. A bolt having one end bent like a hook holds the lug from the beam. In Fig. 44 the lug is replaced by a loop of wrought iron. Fig. 45 shows another method of suspension; the connection between the

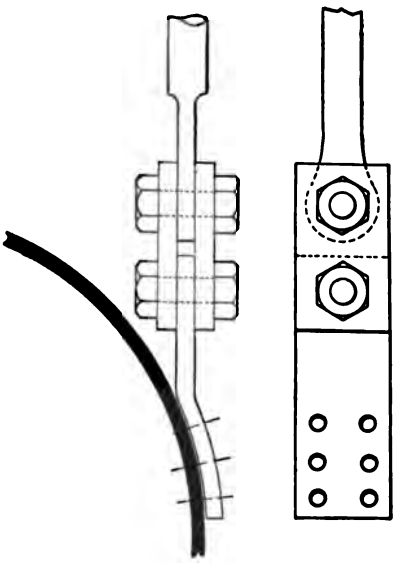


Fig. 45.

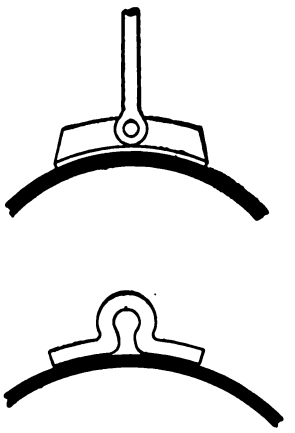


Fig. 46.

rod and boiler plates being short pieces of boiler plate arranged for flexibility.

When the boiler is of small diameter, it may be suspended as shown in Fig. 46.

MANHOLES AND HANDHOLES.

A manhole allows access to the boiler for cleaning and repairs. It is usually elliptical in form and large enough to admit a man. About 16 inches for the major axis and 12 for the minor axis is a good size. The manhole is closed by a plate or cover made of cast or wrought iron. This plate is held to the seat by a yoke or yokes, and bolts. Fig. 47 shows one form, Y being the yoke, L the cover and N the bolt. The joint between the cover and the shell is made steam tight by packing.

The **strength** of the manhole is important. Wherever a

large hole is cut in the plate the edge should be strengthened, for the tension is concentrated there and the plates are, moreover, likely to become weak by corrosion. The *strain* put upon the plate by screwing up the cover is considerable, especially if a piece of scale gets between the faces, and the joint is then made tight.

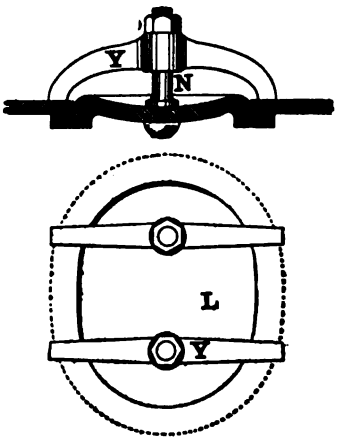


Fig. 47.

Fig. 48 shows the section of a strong and simple manhole. The edge of the plate is strengthened by a broad ring of steel. The ring is flanged and riveted to the shell, its edge forming the seat. The cover as shown in the figure is shaped for strength. The edge of the ring which forms the seat and the cover are machined to make a tight joint without packing. The strengthening ring should be at least $\frac{5}{8}$ of an inch thick and 4 inches wide, that the rivet holes may not be too near the edge. The rivets should have a pitch of about 3 inches.

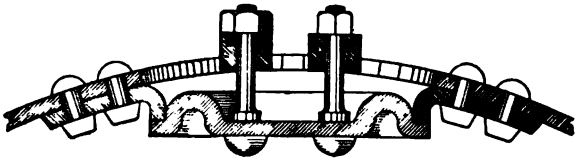


Fig. 48.

Handholes and **mudholes** are more commonly placed in boilers, which are so constructed that a man cannot enter; in a vertical boiler for example. They are used to some extent in other boilers; in horizontal return tube boilers there is usually a handhole in each end near the bottom. Handholes are very convenient to admit hose for washing out the boiler, also for removing scale and sediment. Handholes are similar to manholes in construction, but require only one yoke and one bolt to keep them in place. Mudholes should be provided in order that the sediment and detached scale may be removed without lifting it to the top manhole. Mudholes and handholes greatly facilitate cleaning the fire-box water leg of locomotive and small vertical boilers.

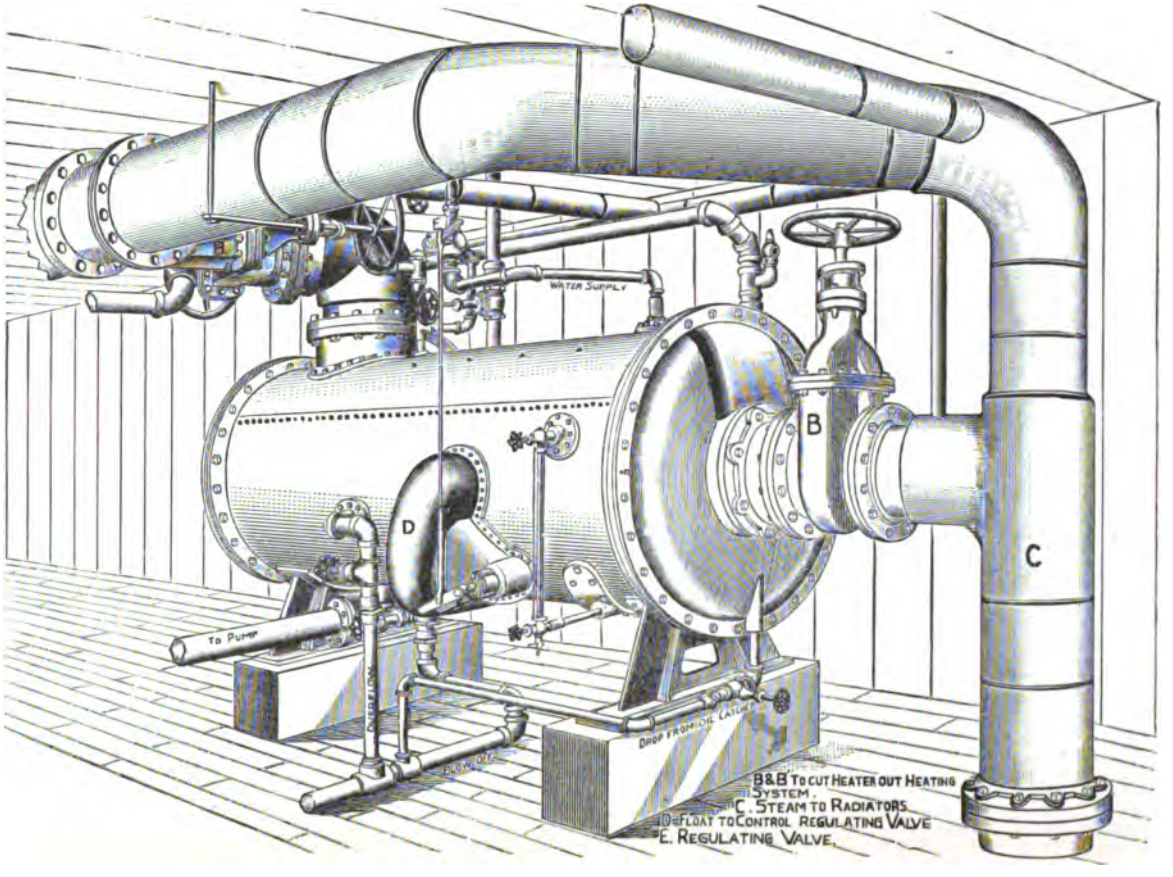
FEED APPARATUS.

There is a great difference of opinion as to where the feed water should be introduced into a boiler; although in common practice it is admitted at the rear, near the bottom. The best place depends upon the arrangement of the heating surfaces. Whatever may be the saving by introducing it at some particular point, there are practical considerations that must not be overlooked. If the entering water is of high temperature, it makes less difference where it is introduced.

In externally fired boilers, the feed pipe is sometimes introduced at the top and extended downward to within a few inches of the bottom. If the water is cold and the plates near which it is delivered are hot, this practice is dangerous, as the sudden chilling of the hot plates causes leaking, rupture, etc. If the pipe is cut off so that there is about two feet between the bottom of the boiler and the end of the pipe, the injurious effects are diminished. Another plan is to fix a vessel under the end of the feed pipe. Water is better distributed as it flows over the top of the vessel, which may also be used as a mud drum or sediment collector. It is a good plan to admit the feed water by a horizontal pipe, entered at either tube plate a few inches below the low water level, and terminated in a trough or a perforated pipe of large diameter. This method distributes the water and allows it to become warmed before it reaches the bottom.

In internally fired boilers of the Cornish and Lancashire type, the feed is usually delivered near the bottom by a horizontal pipe through the front end plate, or by a vertical pipe through the crown. If the feed water is admitted near the bottom, it has a tendency to remain there and increase the amount of dead water that remains cold because of poor circulation.

By introducing it at a high point the circulation is improved since the cold water, being heavier, decends. Temperatures also, are equalized by the heating of the water during its fall. There is another reason why it is advisable to introduce the feed water a few inches below the low water level. If a chip or piece of scale lodges on the seat of the check valve and it fails to work, the boiler may be drained should the feed pipe enter near the bottom.



STILWELL HEATER, PURIFIER AND RECEIVING TANK—Reibold Building, Dayton, Ohio

By making the internal feed pipe of considerable length, the water is warmed before it mixes with the hot water in the boiler.

In internally fired vertical boilers, the feed pipe is short and enters just below the water level.

With feed water containing a large per cent. of bi-carbonate of lime, the feed pipe should terminate in a trough, or a pipe of large diameter with the upper side cut away. If a perforated pipe is used, the small holes are likely to be clogged by the rapid precipitation of lime.

The feed supply should always be regulated to keep the water level as nearly stationary as possible. It is more economical and far better for the boiler, than to wait for the water level to fall and then feed a few inches rapidly. The sudden introduction of a large volume of water causes contraction of the plates, leakage and necessitates rapid firing.

The introduction of feed water at a high temperature increases the economy and tends to prolong the life of the boiler. The increase in economy may be shown by a few figures.

Suppose the steam pressure in a certain boiler is 100 pounds by the gage and the feed water enters at 60° F. What will be the per cent gain if the feed is heated to 210° F.?

The number of heat units required to convert a pound of water at a given temperature, into steam at a given pressure, is equal to the difference in the total heat of the water and steam. From the definition of B. T. U. we see that the total heat in one pound of water at 60° F. is equal to $60 - 32 = 28$ B. T. U.; 32° being the freezing point. The total heat in one pound of steam at 115 pounds absolute pressure is 1,185 (see 3d column of steam tables, page 110.) The difference between these two total heats is $1{,}185 - 28 = 1{,}157$ B. T. U.

In using the steam tables always convert gage pressure into absolute pressure by adding 15 pounds. Hence, the absolute pressure of 100 pounds (gage) is 115 pounds.

If the water is heated from 60° F. to 210° F., the number of B. T. U. gained by heating is $210 - 60 = 150$. The gain in per cent would be,

$$\frac{150}{1157} = .18 \text{ (nearly) or 18 per cent.}$$

The injurious effects from unequal expansion are diminished, and when the feed is warmed by exhaust steam or the waste gases, the saving of fuel is considerable.

There are several ways of heating the feed water. In condensing engines, the hot water from the hot well is used; the temperature in the well being from 100° to 140° F. With a non-condensing engine, the feed water may be heated by the exhaust

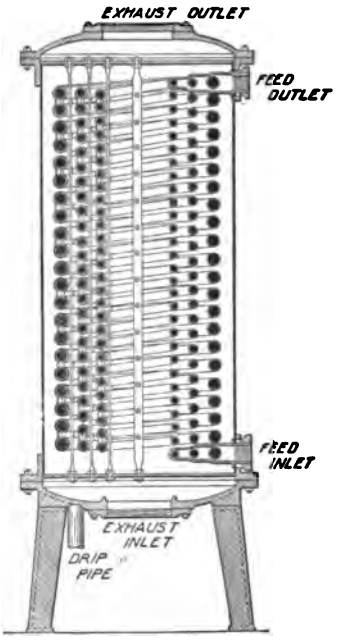


Fig. 49.

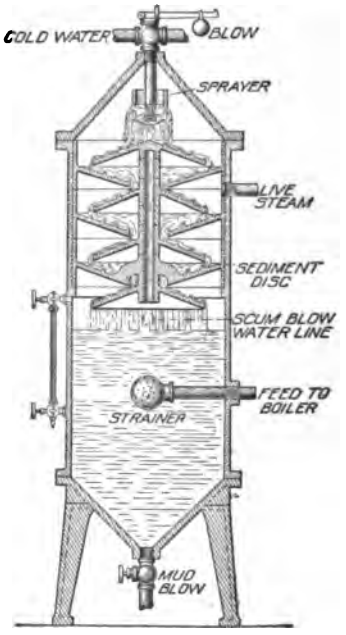


Fig. 50.

steam, or by the waste gases from the chimney. The apparatus used to heat feed water by exhaust steam are called feed water heaters; those which utilize waste gases are called economizers.

There are two types of **heaters.** The **open** and the **closed.** In the open heater the steam raises the temperature of the water by mingling with it, that is, by direct contact. The closed type of heater resembles a surface condenser; the exhaust steam surrounds copper or brass tubes which contain water, or the water circulates about tubes through which steam passes. Fig. 49 shows

a feed water heater, of the closed type, the exhaust steam heating the feed water within the tubes. The heater shown in Fig. 50, is of the open type, the feed water becoming heated and depositing sediment while flowing from one tray to another.

The Cochrane heater, Fig. 51, is a combined heater and purifier. It is of the open heater type. The water enters at the top and flows in a thin sheet over a series of trays. The exhaust steam enters through the oil separator and rises among the trays, heating the water to about 210°F. The action is similar to that of a jet condenser. The gases in the feed water are liberated by the heat and escape into the atmosphere. The mineral impurities in solution, which cause scale, are precipitated by the heat and are deposited on the trays instead of on the plates and tubes of the boiler. The impurities, mud, clay, etc., which fall while the water is at rest, settle because of the large surface and consequent slight velocity. Coke, hay, etc., are used for filters; the heater being constructed so that the hay or coke will not enter the pump. The impurities having less specific gravity than the water, collect at the surface and are removed by flushing. Some of the advantages claimed for this heater are: *1*. The amount of cold water that can be heated is limited only by the quantity of exhaust steam. *2*. All the condensed exhaust is used because the oil is separated. *3*. It can be constructed of cast iron, which is cheap, durable, rarely pits, and resists corrosion. *4*. It is not likely

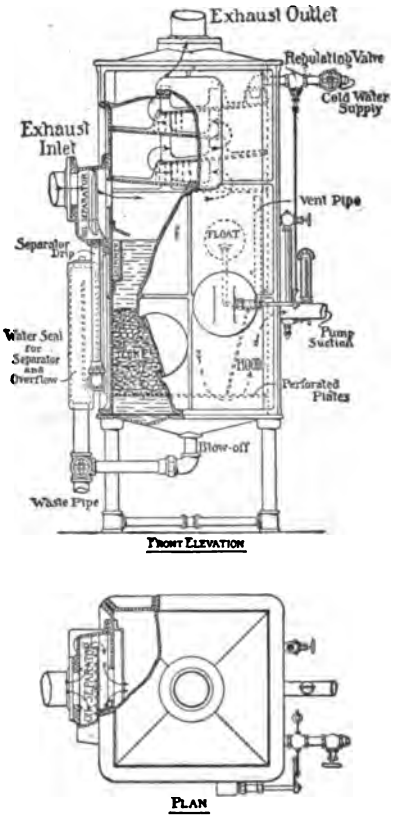


Fig. 51.

to crack or leak, because the internal pressure is rarely more than one pound above atmospheric pressure.

PUMPS AND INJECTORS.

Every boiler should have a reserve apparatus for pumping, or injecting the feed water.

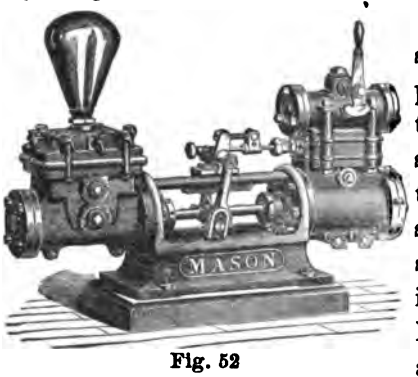


Fig. 52

Boilers are usually fed by a small direct-acting **steam pump** placed near the boiler. Although these pumps require a large steam supply per horse-power per hour, the total amount of steam used is small because the work done is small. A more economical pump is the power pump driven by the large steam engine; but the rate at which water is supplied is not easily regulated to the demand of the boiler. They are usually arranged to pump a larger quantity of water than is required; the excess being allowed to flow back into the suction pipe through a relief valve. The pump shown in Fig. 52 is well adapted for feeding boilers. In Fig. 53 is shown the section of a duplex Worthington steam pump. The action of each is similar. Steam controlled by valves drives the piston in the steam cylinder, which moves the plunger in the water cylinder, since both are fastened to the same rod. The movement of the plunger forces a part of the water in front of it up through the valves into the air chamber and through the pipes into the boiler. On account of the partial vacuum caused by the movement of the plunger, water will be drawn from the suction

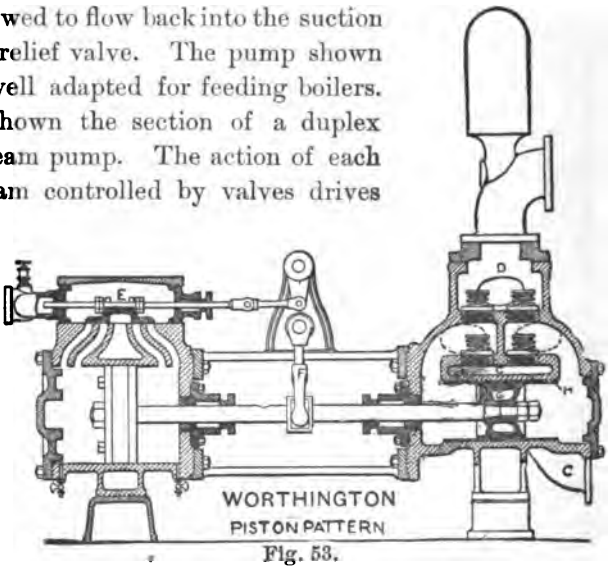


Fig. 53.

pipe through the valves into the pump cylinder, filling the space left by the movement of the plunger. During the return stroke, this water is forced up into the air chamber and a like quantity enters the other end of the pump cylinder. The valves are kept on the seats by light springs, until the pressure on the bottom side is sufficient to lift them and allow water to flow through. When two pumps are placed side by side, and have a common delivery pipe, the machine is called a duplex pump. It is usual to set the steam valves so that when one piston is at the end, the other is at the middle of its stroke. A duplex pump having a large air chamber and valves set to act in this manner delivers water with an approximately constant velocity.

Water may be forced into a boiler by **injectors**, or **inspirators.** By means of this instrument, the energy of a jet of steam is used to force the water into the boiler. That there is sufficient energy to do this work is evident from the fact that each pound of steam in condensing gives up 966 B. T. U. and a B. T. U. is equivalent to 778 foot-pounds. All the energy of the jet of steam is not used in forcing water into the boiler; some is wasted and some is used to heat the feed water.

The action of the injector is briefly as follows. The steam escapes from the boiler with great velocity, and as it passes through the cone-shaped portion, draws air along with it and thus creates a partial vacuum in the suction pipe. Atmospheric pressure forces water up into the suction pipe, and the jet of steam which it meets is partly condensed. The energy of the jet carries the water along with it into the boiler.

Experiments show that the injector if considered as a pump has a very low efficiency. When used for feeding a boiler it has a thermal efficiency of nearly 100 per cent., since all the heat of the steam passes to the water except the slight amount lost in radiation. The pump, however, has one great advantage over the injector; it can force hot water from a heater into the boiler, while an injector can be used only with cold or moderately warm water.

Figs. 54 and 55 show the interior section and exterior of a Hancock Inspirator. To inject water to the boiler, first open Overflow valves 1 and 3, close valve 2 and open Starting valve in the steam pipe. When the water appears at the Over-

flow close 1, open 2 one-quarter turn and then close 3. The inspirator will then be in operation. When the inspirator is not working, open both 1 and 3 to allow water to drain from it.

Both temperature and quantity of delivery water can be

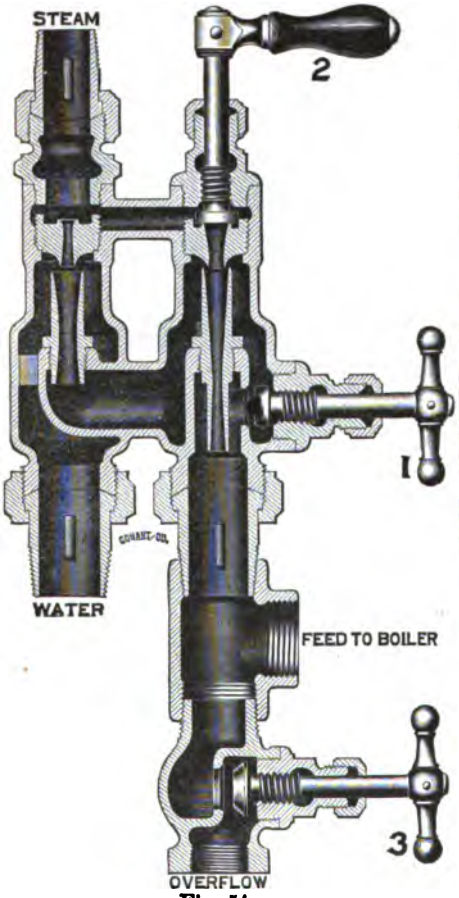


Fig. 54.

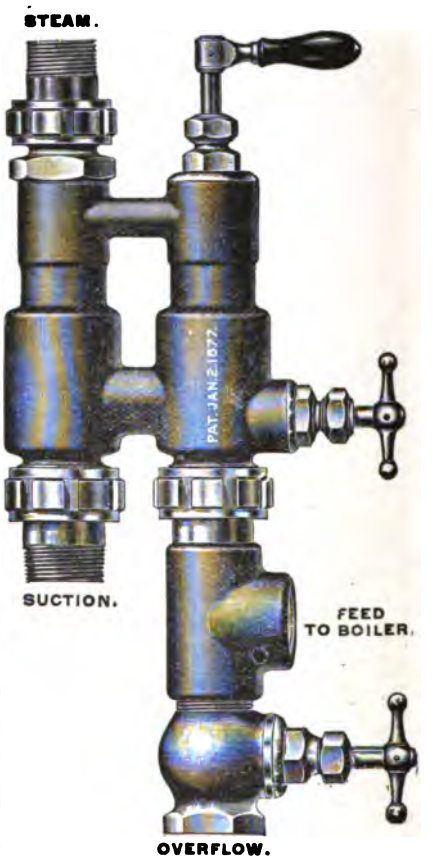


Fig. 55.

varied by increasing or decreasing the water supply. When the water in the suction pipe is hot, either cool off both pipe and injector with cold water, or pump out the hot water by opening and closing the Starting valve suddenly. The following table gives some dimensions and capacities of the different sizes.

THE HANCOCK INSPIRATOR.

Size. (number)	Capacities per Hour.		Pipe Connections.			
	With 60 lbs. Steam Pressure and 4 ft. Lift.	Maximum Horse Power.	Steam.	Suction.	Delivery.	Overflow.
7½	60 gals.	8	⅜	⅜	⅜	¼
8¾	90 "	12	⅜	½	½	⅜
10	120 "	16	⅜	½	½	⅜
12½	220 "	30	½	¾	¾	½
15	300 "	40	½	¾	¾	½
17½	420 "	56	¾	1	1	¾
20	540 "	72	¾	1	1	¾
22½	720 "	96	1	1¼	1¼	1
25	900 "	120	1	1¼	1¼	1
30	1,260 "	168	1¼	1½	1½	1¼
35	1,740 "	230	1¼	1½	1½	1¼
40	2,230 "	298	1½	2	2	1½
45	2,820 "	376	1½	2	2	1½
50	3,480 "	464	2	2½	2½	2
55	3,650 "	600	2	2½	2½	2

PIPES AND PIPE COVERINGS.

Piping a steam plant must be carefully done. The pipes should be large enough, well lagged, and so placed and arranged that there will be slight steam friction and as little condensation as possible. The steam pipe leading from the boiler called the main steam pipe or "log," is usually made of cast iron. It is made in sections flanged and bolted together.

Changes of temperature cause expansion and contraction of the steam pipes, and if no provision is made for expansion, the pipes or fittings may crack, or the joints become loosened and leak. If a pipe is long and straight, some form of expansion joint is generally used. A bend or elbow will allow the pipe to expand without injury to itself or boiler. There are various other methods; the use of a curved copper pipe shaped like an inverted U; expansion joints, etc. If a slip joint is employed it should be in line to prevent its becoming cramped.

If high pressures are used, steel pipe is better than cast iron, for it is much stronger, more flexible and reliable.

If a stop valve is placed in a vertical pipe, the portion of the

pipe above the valve will fill up with water when the valve is closed. In order that the water may not be blown over to the engine a drip pipe or a steam trap should be used. The pipe which supplies an engine, usually has a drip pipe above the throttle valve.

In case horizontal pipes are arranged so that water may collect in them, any increase in the velocity or the sudden opening of a valve will sweep the water along with the steam. This water causes considerable damage to the engine and also to the pipe. The sudden rush of water against elbows is likely to cause leakage if not actual bursting.

The true inside diameter of a steam, gas or water pipe is not always the same as the name of the pipe indicates. For instance; what is called a 3-inch pipe has an actual inside diameter of 3.067 inches and an outside diameter of 3.5.

The following table gives the principal dimensions of steam pipes.

Standard Sizes, etc., of Wrought Iron Pipe for Water, Gas, or Steam.

(Briggs Standard.)

Diameter of Tube.			Thickness of Metal.	Internal Circumference.	External Circumference.	Internal Area.	External Area.
Nominal Inside.	Actual Inside.	Actual Outside.					
INS.	INS.	INS.	INS.	INS.	INS.	INS.	INS.
⅛	.270	.405	.068	.848	1.272	.0572	.129
¼	.364	.540	.088	1.144	1.696	.1041	.229
⅜	.494	.675	.091	1.552	2.121	.1916	.358
½	.623	.840	.109	1.957	2.652	.3048	.554
¾	.824	1.050	.113	2.589	3.299	.5333	.866
1	1.048	1.315	.134	3.292	4.134	.8627	1.357
1¼	1.380	1.660	.140	4.335	5.215	1.496	2.164
1½	1.610	1.900	.145	5.061	5.969	2.038	2.835
2	2.067	2.375	.154	6.494	7.461	3.355	4.430
2½	2.468	2.875	.204	7.754	9.032	4.783	6.491
3	3.067	3.500	.217	9.636	10.996	7.388	9.621
3½	3.548	4.000	.226	11.146	12.566	9.887	12.566
4	4.026	4.500	.237	12.648	14.137	12.730	15.904
4½	4.508	5.000	.246	14.153	15.708	15.939	19.635
5	5.045	5.563	.259	15.849	17.475	19.990	24.299
6	6.065	6.625	.280	19.054	20.813	28.889	34.471
7	7.023	7.625	.301	22.063	23.954	38.737	45.663
8	7.982	8.625	.322	25.076	27.096	50.039	58.426
10	10.019	10.750	.366	31.475	33.772	78.838	90.762

If the area of a steam pipe is small there will be considerable loss in pressure due to friction. It is usual to limit the velocity of live steam in pipes to 5,000 or 6,000 feet per minute. If there are many elbows or bends the allowable velocity may be reduced to 4,500 feet or less.

Suppose we wish to find the **diameter** of the **main steam pipe** from the boilers to the engines. The engines furnish 4,500 horse-power and use 20 pounds of steam per horse-power per hour. Also the pressure is 105 pounds absolute, that is, about 90 pounds by gage.

The total amount of steam furnished per minute will be,

$$4{,}500 \times 20 \div 60 = 1{,}500 \text{ pounds.}$$

Since one pound of steam at 105 pounds absolute pressure has a volume of 4.205 cubic feet, the total volume of steam passing through the pipe per minute will be $1{,}500 \times 4.205 = 6{,}307.5$ cubic feet. Then if we allow a velocity of 5,000 feet per minute the area will be $\frac{6307.5}{5000} = 1.26$ square feet (approx.) If the pipe has 1.26 square feet or 181 inches of area it would be 15.2 + inches in diameter. The pipe used would be the next size, that is, 16 inches.

The **thickness** of pipes has been discussed in Mechanics. For copper pipes Unwin gives the following formula:

$$t = .0001\ p\ d + \tfrac{1}{8},$$

in which t is the thickness in inches, p the steam gage pressure, d the diameter and $\frac{1}{8}$ a constant.

To find the thickness of a copper pipe $3\frac{1}{2}$ inches in diameter, with a steam pressure of 80 pounds; $t = .0001\ p\ d + \frac{1}{8}$ $= .0001 \times 80 \times 3\frac{1}{2} + \frac{1}{8} = .153$ inch.

The sections of the main steam pipe are bolted together at the flanges. The joint is made tight by machining the flanges and inserting between them a packing ring of India rubber, asbestos, or gutta percha. Sometimes a corrugated copper ring is used in place of the rubber, or a groove cut in the flanges, in which is fitted a copper wire.

Water in steam pipes causes **water hammer**. For this reason the pipes should be arranged or provided with drip pipes, traps, separators, etc., so that the pipes will not be injured.

In order to return the water of condensation from a drip, trap, or separator to the boiler, a **steam loop** is used. It consists of a

riser which extends vertically 10 or 12 feet and joins a horizontal. Another vertical pipe called the **drop leg** connects this horizontal with the boiler. If the hydrostatic pressure in the drop leg is sufficient to overcome the difference in pressure between the separator and the boiler, water of condensation flows into the boiler.

When steam pipes are exposed to open air, the amount of steam that will condense in them depends upon the condition of the surfaces, and the temperatures and velocities of the air and steam. This pipe condensation is magnified by, and adds considerable to, the cylinder condensation. Tests show that from 2 to 2.8 B. T. U. are lost per square foot of bare pipe, per hour, per degree difference in temperature. In winter each 10 square feet of uncovered pipe will condense about 100 pounds of steam per day. When we come to consider the amount of steam lost per year from a long pipe it is evident that if this wasted heat could be saved it would mean considerable money for the owner.

To make this loss as small as possible it is customary to clothe or cover the pipe with some material that will prevent loss and will not burn. There is considerable difference in the value of various substances when used to prevent radiation of heat from pipes. The value varies nearly in the inverse ratio of the conducting power. The following table of the relative value of various substances to prevent radiation has been compiled from several sources.

Felt, Hair or Wool	1.00
Mineral wool	.68—.83
Straw rope, spiral	.77
Carbonate of magnesia	.67—.76
Charcoal	.63
Sawdust	.61—.68
Wood ashes	.61
Wood	.40—.55
Loam	.55
Chalk	.50
Gas-house Carbon	.47
Asbestos Paper	.47
Asbestos, fibrous	.36
Plaster of Paris	.34
Coke lumps	.27
Air space (undivided)	.22
Sand	.17
Glass	.05

The best protection is air confined in minute cells. The value of different substances depends upon this fact. Air space alone is one of the poorest of non-conductors. There are several good non-conductors, hair or wool felt, for instance, but they soon become charred if high pressure is used. Mineral wool, a fibrous material made from blast furnace slag, is a good non-combustible material, but is brittle and likely to fall off if much jarring exists.

A good covering is made by wrapping three or more layers of asbestos paper, each layer being about $\frac{1}{16}$ inch thick, around the pipe and covering these layers with a layer of hair about $\frac{3}{4}$ inch thick, the whole being wrapped in canvas or paper. This covering will last for about five years with high pressure steam, and for ten or fifteen, with low pressure. Another method: wrap the pipe in asbestos paper, then lay strips of wood lengthwise of the pipe and bind with wire. Wrap roofing paper around the wood frame, fastening with paste, wire or twine.

Coverings should not be less than one inch thick; when much thicker the increase of economy is so slight that it doesn't pay the first extra cost. There are a large number of coverings manufactured to fit various sizes of pipes. They are usually made in sections and can be readily put on or detached. It is of great importance that the pipe covering should be kept in repair, for a loosely hanging covering is of little value.

The tops and the portions of the surfaces of boilers which are in contact with the hot gases should be covered with some good non-conducting material to prevent loss of heat by radiation.

Vertical boilers are often covered with layers of asbestos or magnesia mortar. It is applied with a trowel in layers; the thickness varying with the steam pressure.

CALORIMETERS.

Steam from a boiler is generally accompanied by more or less water or, as usually expressed, moisture. In order to determine the quantity of heat supplied with this mixture it is necessary to ascertain the relative amounts of these. This is accomplished by an instrument called a calorimeter.

The Barrel Calorimeter was invented by the distinguished Alsatian engineer, Mr. G. A. Hirn. It is one of the earliest forms and is used only when more accurate and expensive apparatus cannot be obtained.

As many corrections are required, it is recommended only for approximate results. It consists of a barrel holding at least 400 pounds of water, and an iron pipe having a valve. This pipe is connected with the source from which steam is to be taken, and the end has a hose provided at its extremity with a cap closed at the end, but perforated on its side with a number of holes. A propeller is placed in the barrel for stirring the water, so as to obtain as nearly as possible a uniform temperature. The whole is then placed upon platform scales, as illustrated in Fig. 56. To use, the barrel is first filled about two-thirds with cold water, carefully weighed, and its temperature noted. Before immersing the hose in the barrel, steam is allowed to flow through the pipe and hose until they are thoroughly warmed. The hose is then plunged into the barrel and steam allowed to flow until a sufficient amount is obtained; the valve is then closed, the hose removed, and the resulting mixture weighed and temperature noted. The calculation is made by the use of the following formula:

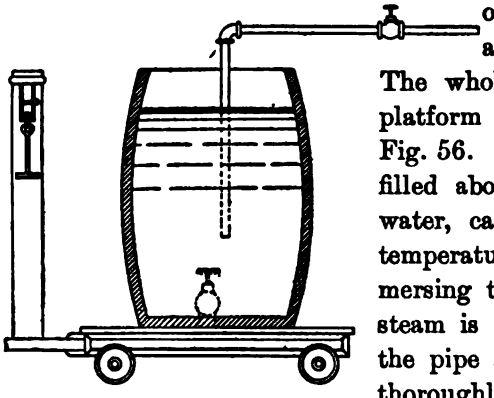


Fig. 56.

$$x = \frac{w\,(q - q_1) - \mathrm{W}\,(q - q_2)}{(\mathrm{W} - w)\,r}$$

in which,

x = the weight of steam contained in one pound of the mixture from the boiler or other source,

w = the weight of the cooling or condensing water,

W = the final weight of the water in the barrel,

q = the heat of the liquid mixed with the steam,

q_1 = the heat of one pound of the cooling water before the mixture from the boiler is added,

q_2 = the heat of one pound of the water after the mixture from the boiler is added,

*r = the heat of evaporation corresponding to the absolute pressure at the source of supply.

If x is greater than 1 it shows that the steam is superheated, the number of degrees of superheating being equal to

$$2.0833\ r\ (x - 1).$$

Example: Let $w = 455$ pounds, $W = 495$ pounds,

$q = 276.9$, corresponding to an absolute pressure of 75 pounds,

$r = 898.8$ the corresponding heat of evaporation,

$q_1 = 18.10$, corresponding to a temperature of 50° F., the initial temperature of the cooling water,

$q_2 = 108.2$, corresponding to a temperature of 140° F., the final temperature of the water in the barrel, then:

$$x = \frac{455\ (276.9 - 18.10) - 495\ (276.9 - 108.2)}{(495 - 455) \times 898.8}$$

$= 0.952$ pound or 95.2 per cent, or the moisture $= 1 - 0.952 = 0.048$ of a pound of 4.8 per cent.

A slight error either in weighing or in taking the temperature may cause a large error in the result. The above formula is based upon the assumption that there is no loss from radiation.

Separator Calorimeter.† This instrument consists of a chamber, A, Fig. 57, about a foot long and three inches in diameter, into which is led a steam-pipe D from the boiler, the extension of this pipe being enlarged within the chamber A. The lower end of the pipe D is perforated with a number of holes, each $\frac{1}{8}$ inch in diameter. The steam escapes from the calorimeter through the pipe S, in which is secured a diaphragm having a hole in its center $\frac{1}{16}$ inch in diameter. On entering the calorimeter by the pipe D the velocity of the steam is reduced, owing to the enlargement, and the particles of water are carried to the bottom of the chamber while the steam rises to the top, escaping at S, as mentioned. A water-glass G is attached, as shown. The height of the water in the glass at the beginning is noted and marked by a string or other means, and the water is again brought to the same

*Note.—Some authors use t with suffixes, instead of q with suffixes, L instead of r, and Q instead of x.

† The invention of Professor R. C. Carpenter.

level at the end of the test, by opening the cock E, and the amount drained very accurately weighed. The results may be calculated by aid of the following formula:

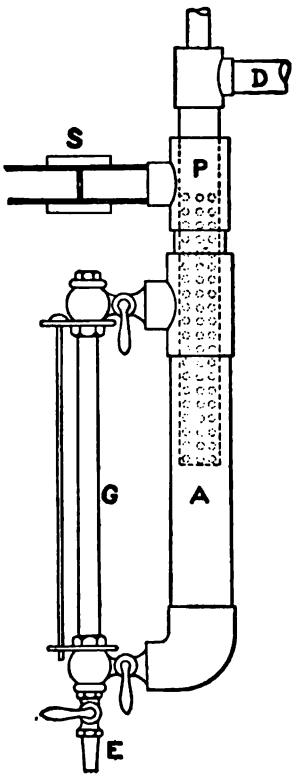


Fig. 57.

$$x = \frac{w_1 + W}{W + w}$$

where

x = the weight of steam contained in one pound of the mixture,

W = weight of steam discharged from the pipe S,

w = weight of water drawn through the cock E,

w_1 = weight of steam condensed by loss of heat due to radiation.

The loss w_1 may be determined by uniting two instruments, the first one discharging its steam through S, in which case no diaphragm is used, into the steam-pipe D of the second, the amount of water collected in the second being that due to the loss by radiation, since the first supplies the second with steam only.

Example:

W = 10 pounds,

w = 6 ounces, = $\frac{3}{8}$ of a pound,

w^1 = 2 ounces, = $\frac{1}{8}$ of a pound,

from which we obtain

$$x = \frac{\frac{1}{8} + 10}{10 + \frac{3}{8}} = 0.9759 \text{ of a pound, or } 97.59 \text{ per cent.}$$

The moisture is then:

1 — 0.9759 or 0.0241 of a pound, or 2.41 per cent.

In using this instrument gages and thermometers are not required, as it depends upon weighing the steam and water.

The Throttling Calorimeter was invented by Professor Cecil H. Peabody and is made with varying constructive details.

Fig. 58 shows the general arrangement. The mixture of steam and water from the boiler is taken from the main steam-

pipe through what is termed a sampling pipe. Various forms of this pipe are made; one arrangement consists of a pipe closed at its inner end, but having numerous holes $\frac{1}{4}$ inch in diameter drilled staggering around the sides. The calorimeter should be placed as close as possible to the main steam pipe and the gage for indicating the pressure in the main steam pipe on the latter and near the calorimeter. The gage is sometimes connected to a tee on the pipe leading to the calorimeter, but it is better to have this gage where the velocity of the flowing steam is less. A valve is placed in the pipe to the calorimeter, below which is inserted a nipple, A, having a small converging orifice, D, about two-tenths of an inch in diameter and very carefully made. The object of such an orifice is to determine the weight of steam flowing through the calorimeter so an allowance can be made for the loss when testing an engine, or boiler, where the net weight used is required. A cup, B, is screwed into the top for holding an accurate thermometer. The cup is made of brass and is filled with oil, but if mercury is used the cup must be of iron or steel. A delicate gage, C, for determining the pressure in the calorimeter and a pipe and valves at the bottom complete the apparatus. The valve N is sometimes omitted and a simple pipe used, as the throttling is best accomplished by use of the valve E, or orifice D. All pipes leading to the calorimeter should be well covered with a good non-conductor. To use the instrument proceed

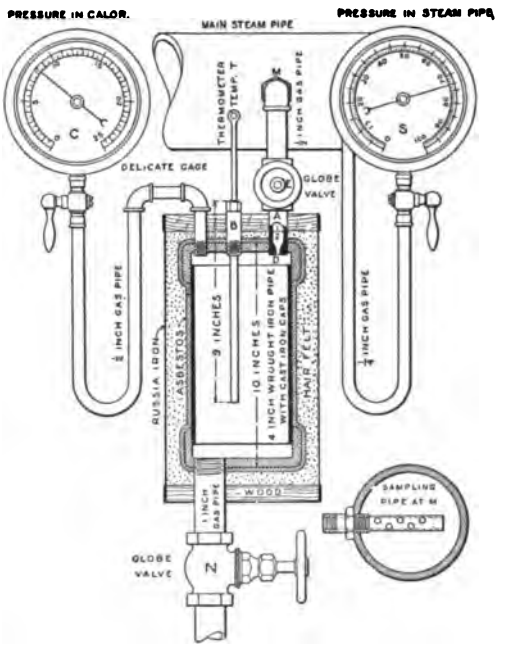


Fig. 58.

as follows: Open wide valves E and N, to bring the apparatus to a uniform temperature, then gradually close E until the steam in the calorimeter is superheated; that is, until the temperature as shown by the thermometer is greater than that corresponding to the absolute pressure determined from the reading of the gage C and barometric pressure. The result may now be calculated by the following formula:

$$x = \frac{\lambda_c + 0.48\ (T - t_c) - q_s}{r_s}$$

in which

x = the weight of steam contained in one pound of the mixture from the main steam pipe or other source,

λ_c = the total heat corresponding to the absolute pressure determined from the reading of the steam gage C and barometric pressure.*

T = the temperature as shown by the thermometer,

t_c = the temperature of steam corresponding to the absolute pressure as determined by the reading of the gage C and barometric pressure,

q_s = the heat of the liquid corresponding to the absolute pressure in the steam pipe,

r_s = the heat of evaporation corresponding to the absolute pressure in the steam pipe,

0.48 = the heat required to superheat the steam one degree Fahrenheit under constant pressure.

Example:

Barometric pressure 14.78 lbs. Absolute pressure in the main steam pipe 87.78 lbs. Absolute pressure in the calorimeter 23.03 lbs. Temperature T = 260° F.,

Then

$\lambda_c = 1153.68$ $\qquad$ $q_s = 288.1$

$t_c = 235.28$ $\qquad$ $r_s = 890.88$

and

$$x = \frac{1153.68 + .48(260 - 235.28) - 288.1}{890.88}$$

= 0.984 of a pound, or 98.4 per cent of the mixture is steam, or the moisture = (1 − 0.984) = 0.016 of a pound, or 1.4 per cent.

*Some steam tables use H instead of the Greek letter λ (lambda).

This form of calorimeter is suitable only for cases where the moisture does not exceed three per cent of the mixture. Its principle is based upon the assumption that there is no loss of heat, in which case steam mixed with a small amount of water is superheated when the pressure is reduced by throttling.

HORSE-POWER OF BOILERS.

The unit which we call the horse-power is *arbitrary*. Assuming that 30 pounds of steam are required per horse-power per hour for an average engine, this unit for boilers has been adopted.

The unit of horse-power is the evaporation of 30 pounds of water from a temperature of 100°F. into steam at 70 pounds gage pressure which is considered equivalent to the evaporation of $34\frac{1}{2}$ pounds from and at 212° F. A horse-power is equal to 33,305 B. T. U. per hour.

As all boilers do not generate steam at the same pressure and from the same temperature of feed water, it is necessary to reduce the **actual** evaporation to an **equivalent** evaporation. Unless this is done their relative performances cannot be compared.

For this comparison, the actual evaporation is reduced to an evaporation at 70 pounds gage pressure, the feed water being at 212°F. and the fuel without ash.

Let W = the water actually evaporated in pounds,
H = the total heat of steam above 32°F., at actual pressure,
T = temperature of feed water,
w = equivalent evaporation from and at 212°F.:

Since 966 B. T. U. are necessary to evaporate one pound of water from and at 212°F., the equivalent evaporation may be found from the formula,

$$W\ (H + 32 - T) = 966w, \text{ or } w = \frac{W\ (H + 32 - T)}{966}.$$

Then the horse-power of the boiler is,

$$\text{H. P.} = \frac{w}{34.5}.$$

The above method is considerably shortened by substituting for the quantity $\frac{H + 32 - T}{966}$, the number found in the following table which corrresponds to the actual feed water temperature and steam pressure.

FACTORS OF EVAPORATION.

Temperature Feed-Water. Fahrenheit.	PRESSURE IN POUNDS PER SQUARE INCH ABOVE THE ATMOSPHERE. (GAGE PRESSURE.)																				Temperature Feed-Water. Fahrenheit.
	0	5.	15.	25.	35.	45.	55.	65.	75.	85.	95.	105.	115.	125.	135.	145.	155.	165.	175.	185.	200.	
Degr's																						Degr's
32	1.187	1.192	1.199	1.204	1.209	1.212	1.216	1.218	1.221	1.223	1.226	1.228	1.230	1.231	1.233	1.235	1.236	1.238	1.239	1.240	1.241	32
35	1.184	1.189	1.196	1.201	1.206	1.209	1.213	1.215	1.218	1.220	1.223	1.225	1.227	1.228	1.230	1.232	1.233	1.235	1.236	1.237	1.238	35
40	1.179	1.184	1.191	1.196	1.201	1.204	1.208	1.219	1.213	1.215	1.218	1.220	1.222	1.223	1.225	1.227	1.228	1.230	1.231	1.232	1.233	40
45	1.173	1.178	1.185	1.190	1.195	1.198	1.202	1.204	1.207	1.209	1.212	1.214	1.216	1.217	1.219	1.221	1.222	1.224	1.225	1.226	1.227	45
50	1.168	1.173	1.180	1.185	1.190	1.193	1.197	1.199	1.202	1.204	1.207	1.209	1.211	1.212	1.214	1.216	1.217	1.219	1.220	1.221	1.222	50
55	1.163	1.168	1.175	1.180	1.185	1.188	1.192	1.194	1.197	1.199	1.202	1 204	1.206	1.207	1.209	1.211	1.212	1.214	1.215	1.216	1.217	55
60	1.158	1.163	1.170	1.175	1.180	1.183	1.187	1.189	1.192	1.194	1.197	1.199	1.201	1.202	1.204	1.206	1.207	1.209	1.210	1.211	1.212	60
65	1.153	1.158	1.165	1.170	1.175	1.178	1.182	1.184	1.187	1.189	1.192	1.194	1.196	1.197	1.199	1.201	1.202	1.204	1.205	1.206	1.207	65
70	1.148	1.153	1.160	1.165	1.170	1.173	1.177	1.179	1.182	1.184	1.187	1.189	1.191	1.192	1.194	1.196	1.197	1.199	1.200	1.201	1.202	70
75	1.143	1.148	1.155	1.160	1.165	1.168	1.172	1 174	1.177	1.179	1.182	1.184	1.186	1.187	1.189	1.191	1.192	1.194	1.195	1.196	1.197	75
80	1.137	1.142	1.149	1.154	1.159	1.162	1.166	1.168	1.171	1.173	1.176	1.178	1.180	1.181	1.183	1.185	1.186	1.188	1.189	1.190	1.191	80
85	1.132	1.137	1.144	1.149	1.154	1.157	1.161	1.163	1.166	1.168	1.171	1.173	1.175	1.176	1.178	1.180	1.181	1.183	1.184	1.185	1.186	85
90	1.127	1.132	1.139	1.144	1.149	1.152	1.156	1.158	1.161	1.163	1.166	1.168	1.170	1.171	1.173	1.175	1.176	1.178	1.179	1.180	1.181	90
95	1.122	1.127	1..34	1.139	1.144	1.147	1.151	1.153	1.156	1.158	1.161	1.163	1.165	1.166	1.168	1.170	1.171	1.173	1.174	1.175	1.176	95
100	1.117	1.122	1.129	1.134	1.139	1.142	1.146	1.148	1.151	1.153	1.156	1.158	1.160	1.161	1.163	1.165	1.166	1.168	1.169	1.170	1.171	100
105	1.111	1.116	1.123	1.128	1.133	1.136	1.140	1.142	1.145	1.147	1.150	1.152	1.154	1.155	1.157	1.159	1.160	1.162	1.163	1.164	1.165	105
110	1.106	1.111	1.118	1.123	1.128	1.131	1.135	1.137	1.140	1.142	1.145	1.147	1.149	1.150	1.152	1.154	1.155	1.157	1.158	1.159	1.160	110
115	1.101	1.106	1.113	1.118	1.123	1.126	1.130	1.132	1.135	1.137	1.140	1.142	1.144	1.145	1.147	1.149	1.150	1.152	1.153	1.154	1.155	115
120	1.096	1.101	1.108	1.113	1.118	1.121	1.125	1.127	1.130	1.132	1.135	1.137	1.139	1.140	1.142	1.144	1.145	1.147	1.148	1.149	1.150	120
125	1.091	1.096	1.103	1.108	1.113	1.116	1.120	1.122	1.125	1.127	1.130	1.132	1.134	1.135	1.137	1.139	1.140	1.142	1.143	1.144	1.145	125
130	1.085	1.090	1.097	1.102	1.107	1.110	1.114	1.116	1.119	1.121	1.124	1.126	1.128	1.129	1.131	1.133	1.134	1.136	1.137	1.138	1.139	130
135	1.080	1.085	1.092	1.097	1.102	1.105	1.109	1.111	1.114	1.116	1.119	1.121	1.123	1.124	1.126	1.128	1.129	1.131	1.132	1.133	1.134	135
140	1.075	1.080	1.087	1.092	1.097	1.100	1.104	1.106	1.109	1.111	1.114	1.116	1.118	1.119	1.121	1.123	1.124	1.126	1.127	1.128	1.129	140
145	1.070	1.075	1.082	1.087	1.092	1.095	1.099	1.101	1.104	1.106	1.109	1.111	1.113	1.114	1.116	1.118	1.119	1.121	1.122	1.123	1.124	145
150	1.065	1.070	1.077	1.082	1.087	1.090	1.094	1.096	1.099	1.101	1.104	1.106	1.108	1.109	1.111	1.113	1.114	1.116	1.117	1.118	1.119	150
155	1.059	1.064	1.071	1.076	1.081	1.084	1.088	1.090	1.094	1.096	1.098	1.100	1.102	1.103	1.105	1.107	1.108	1.110	1.111	1.112	1.113	155
160	1.054	1.059	1.066	1.071	1.076	1.079	1.083	1.085	1.088	1.090	1.093	1.095	1.097	1.098	1.100	1.102	1.103	1.105	1.106	1.107	1.108	160
165	1.049	1.054	1.061	1.066	1.071	1.074	1.078	1.080	1.083	1.085	1.088	1.090	1.092	1.093	1.095	1.097	1.098	1.100	1.101	1.102	1.103	165
170	1.044	1.049	1.056	1 061	1.066	1.069	1.073	1.075	1.078	1.080	1.083	1.085	1.087	1.088	1.090	1.092	1.093	1.095	1.096	1.097	1.098	170
175	1.039	1.044	1.051	1.056	1.061	1.064	1.068	1.070	1.073	1.075	1.078	1.080	1.082	1.083	1.085	1.087	1.088	1.090	1.091	1.092	1.093	175
180	1.033	1.038	1.045	1.050	1.055	1.058	1.062	1.064	1.067	1.069	1.072	1.074	1.076	1.077	1.079	1.081	1.082	1.084	1.085	1.086	1.087	180
185	1.028	1.033	1.040	1.045	1.050	1.053	1.057	1.059	1.062	1.064	1.067	1.069	1.071	1.073	1.074	1.076	1 077	1.079	1.080	1.081	1.082	185
190	1.023	1.028	1.035	1.040	1 045	1.048	1.052	1.054	1.057	1.059	1.062	1.064	1.066	1.067	1.069	1.071	1.072	1.074	1.075	1.076	1.077	190
195	1.018	1.023	1.030	1.035	1 040	1.043	1.047	1.049	1.052	1.054	1.057	1.059	1.061	1.062	1.064	1.066	1.066	1.069	1.070	1.071	1.072	195
200	1.013	1.018	1.025	1.030	1.035	1.038	1.042	1.044	1.047	1.049	1.052	1.054	1.056	1.057	1.059	1.061	1.062	1.064	1.065	1.066	1.067	200
205	1.001	1 012	1.019	1.024	1.029	1.032	1.036	1.038	1.041	1.043	1.046	1.048	1.050	1.051	1.053	1.055	1.056	1.058	1.059	1 060	1.062	205
210	1.002	1 007	1.014	1.019	1.024	1.027	1.031	1.033	1.036	1.038	1.041	1.043	1.045	1.046	1.048	1.050	1.051	1.053	1.054	1.055	1.057	210
212	1.000	1.005	1.012	1.017	1.022	1.025	1.029	1.031	1.034	1.036	1.039	1.041	1.043	1.044	1.046	1.048	1.049	1.051	1.052	1.053	1.056	212

For example : A boiler is required to furnish 2100 pounds of steam per hour. If the gage pressure is 85 pounds and the feed water enters at 50°F., what is the equivalent evaporation and what is the horse-power?

From the table the factor for 85 pounds pressure and 50° F. is 1.204. Then the equivalent evaporation would be $1.204 \times 2100 = 2528.4$ pounds, and $\frac{2528.4}{34.5} = 73$ (approx.) = the H. P.

BOILER SETTING.

The setting for a stationary boiler consists of the foundation and as much of the furnace and flues as are external to the boiler shell. Some internally fired boilers, the Lancashire for instance, have flues in the brick setting. The whole furnace and sometimes the flues, as is the case with the plain cylindrical boiler, are in the setting. Vertical boilers have simply a foundation, and locomotive boilers have no setting since they are supported by the frames of the engines. Marine boilers are usually placed on saddles, which are built into the framing of the vessel.

In setting a boiler there are three principal requisites that should be kept in mind. *1.* A stable support or foundation for the shell. *2.* Properly arranged spaces for both furnace flues and ash-pit. *3.* A covering which shall prevent loss of heat by radiation and that will not keep moisture in contact with the plates.

There are two principal methods for support; by **brackets** riveted to the shell plates and by supporting from **overhead girders** by means of hooks, rings, etc. In any case the supports should be so arranged that each shall bear its proper proportion of the load and at the same time allow for expansion. If the boiler is short, brackets are generally used, while for long, plain cylindrical boilers the girder method is the more common. If a very long, cylindrical boiler is supported only at each end, the great weight between the two supports is likely to cause bending and an excessive strain on the lower plates at the middle.

The first requisite for a setting is a good **foundation.** If the ground is firm and favorable to a solid foundation, the excavation need be only three or four feet below the level. If it is soft, the excavation should be deeper and the extra depth filled in with

broken stone mixed in with cement, gravel, etc. The first courses of the foundation should be large stones laid in cement; upon this stone work the walls are built, either of stone or brick to within about six inches of the floor level and above this, brick should be used.

In determining the **area** of the bed, the weight that can be put on one square foot should be estimated carefully. With ordinary condition of the soil, the weight per square foot should not exceed 2000 pounds.

Sometimes the bed is made of concrete about two feet in thickness. If the soil is very firm a foundation of large stonework about three feet wide is built under the side, middle and end walls.

The **supporting** and **enclosing** walls are built upon this foundation. The outer walls at the sides and rear are double; the space between, usually about two inches, being an air space to prevent loss of heat. Projecting bricks, which extend from the outer until they just touch the inner, allow for expansion without decreasing the strength of the inner wall. Fig. 59 shows a boiler in the brick setting. The boiler is supported by brackets. The front brackets rest on iron plates which are built into the walls and the rear brackets are free to move, being supported by rollers. If the furnace is designed for anthracite coal the distance between the shell and the grate bars is about two feet; for softer coal this distance is increased a few inches. The grates have an incline of a few inches so the bed of coal will be thicker at the rear than at the front; this allows a more even consumption of fuel as the air passes through the fire at the bridge more freely. The furnace, both front and sides, and sometimes portions back of the bridge, are lined with firebrick. The bridge also is generally covered with firebrick. The space between the bridge and the shell is from 6 to 8 inches. The rear walls are built a little higher than the top row of tubes. The fire line should not be carried above the water line; if it is, the intense heat is likely to injure the shell plates. Never expose any part of the boiler not covered by water to the gases from the fire. The side walls are built about the same height as the rear walls. The space at the rear is bridged over and stiffened by T irons. In order to increase the heating surface

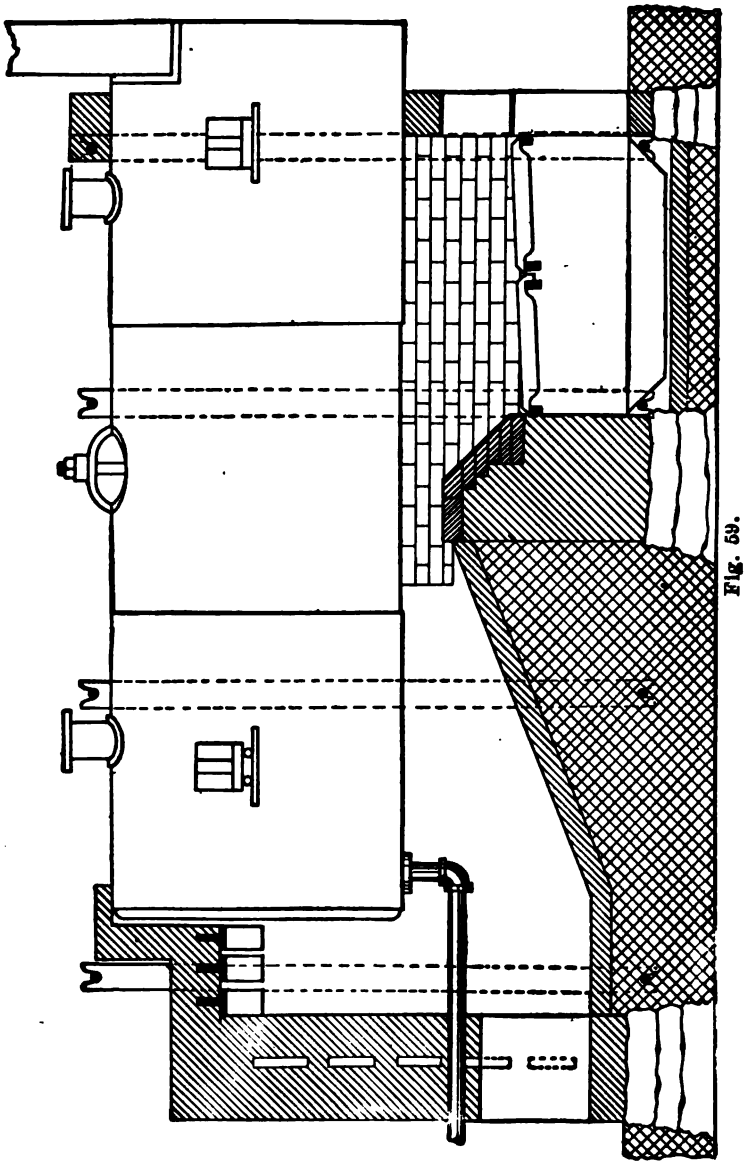


Fig. 69.

the top is arched so that the hot gases will pass over the steam space before they enter the chimney.

The smoke box projects over the front wall and has a rectangular uptake. The side walls are strengthened by buckstaves or binders, which are kept in place by long bolts, secured by nuts on each end, as shown in Fig. 60.

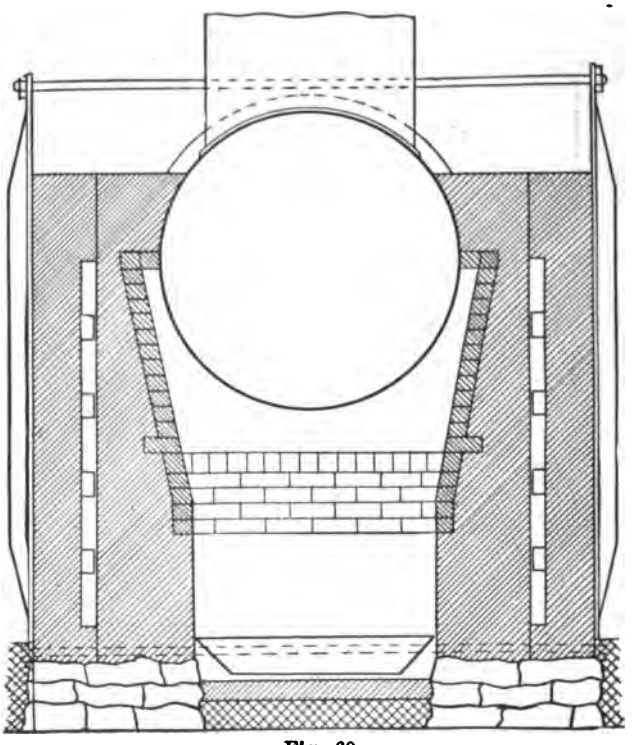


Fig. 60.

Fig. 61 shows the top view of the same boiler.

The front is usually of cast iron with doors for firing, cleaning and for access to the tubes. Soot, dirt, etc., are removed through the door in the brickwork at the rear.

The end which contains the handhole should be set about one inch lower than the other end, so that the sediment and detached scale will tend to accumulate there.

Internally fired boilers may also be enclosed in brickwork. The setting is a support and covering, forming the side flues but

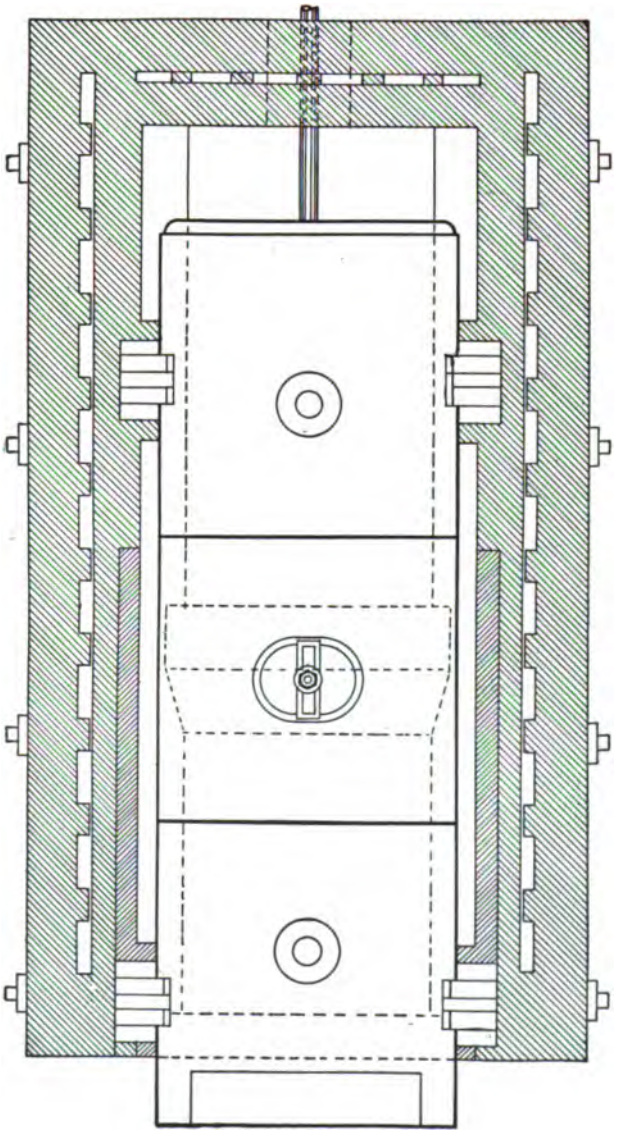


Fig. 61.

not the furnace. Excess of brickwork surface in contact with the shell should be avoided, as brickwork collects moisture which causes external corrosion.

Water Tube Boilers. The settings for water tube boilers are similar to the settings of cylindrical tubular boilers. In "Types of Boilers," the various settings were mentioned. In such boilers as the Babcock and Wilcox, Heine and Stirling, special bridges and tiling are required to conduct the gases and keep them in contact with the tubes. The Cahall boiler is set in a vertical stack, with the furnace at one side.

Marine water tube boilers, as the Almy and Thornycroft are enclosed in sheet iron casing which is lined with non-conducting material; usually asbestos or magnesia.

CORROSION AND INCRUSTATION.

There are several forces which tend to **destroy** and **shorten** the life of every boiler. These forces are divided into two general classes; chemical and mechanical. Impure feed water may injure the boiler in two ways; it may **corrode** the iron, or it may **deposit sediment** that forms scale.

Pure water, free from air and carbon dioxide, has no bad effect on iron; but all natural waters, whether from rain, lake, river, or sea contain air in solution and a little carbon dioxide. Water that contains gases or acids attacks iron readily.

Corrosion. There are two forms of corrosion, **external** and **internal**. External corrosion is caused by exposure to the weather, moisture from the ground, or leakage of joints and fittings. In time a slight leakage from a joint may cause considerable corrosion. A large amount of the deterioration from external corrosion is due to the practice of setting boilers in a mass of brickwork. The brickwork holds any water that may come in contact with the plates. When an internally fired boiler is placed on a wide mid-feather, or supporting wall, all the water from leakage or any other source finds its way to the bottom of the boiler and is kept in contact with the plates by the wall. Mid-feather walls extend beneath the boiler nearly from end to end and are sometimes 18 to 24 inches wide. It is impossible to know from **external appearance**, the condition of the plates resting on the brickwork. The

plates at the edge may appear to be in good condition yet the center portions may be eaten nearly through. If a mid-feather wall is used it should not be more than three or four inches wide, and a few bricks should be loose to permit inspection. When the ground is damp, plates placed between the walls and the shell are exposed to the moisture, thus saving the boiler plates.

Internal corrosion appears in various forms; **general corrosion** or **wasting, pitting** or honeycombing and **grooving.** General corrosion and pitting are the result of chemical action from the feed water, while grooving is a combination of chemical and mechanical action

General corrosion is hard to discover, because it acts uniformly over a large surface. Sometimes the rivet heads waste equally with the plates so that the damage done the plates is not noticeable. Uniform corrosion is the hardest to detect and is the most capricious. Water may attack the plates only at the water line. In some instances the damage is confined to a belt six or eight inches wide at the water level. Sometimes a few rivets or a part of the seams may be corroded, the rest remaining intact. Stays are often weakened more rapidly than boiler plates. The threads of screw stays may be badly corroded and the turned surfaces untouched. The capricious action depends upon the specific gravity of the corrosive agents, the difference of temperature, the difference of structure of the iron, etc. The best way to measure the amount of wasting is to drill the plates. If the thickness of the plates is found to be considerably reduced, the working pressure should be proportionately lowered.

Pitting is clearly defined by the edges of the pits or holes. From this appearance both the extent and depth are well marked. Pitting appears in the form of small holes or patches from $\frac{1}{2}$ inch to 12 inches in diameter, or often as irregular shaped depressions. If the holes are small and close together, the plate is said to be honeycombed. The reason why plates become pitted instead of uniformly corroded has not been satisfactorily explained. It is generally believed to be the lack of homogeneity. Galvanic action between the iron plates and the copper or brass tubes is another claim for the cause, but as pitting occurs when there is no copper or brass near the plates, this theory breaks down.

The phenomenon of pitting is satisfactorily explained as the result of simple chemical action. Acids attack the most susceptible portions of the plate. The corrosion of firebox plates is often considered the result of galvanic action, but there is no doubt that the injury done to plates by drilling and punching is to some extent responsible.

The cause for **grooving**, channeling or furrowing, is not always apparent. It usually is due to the straining, springing, or buckling of the plates, aided by local corrosion. The straining of the plates may be due to insufficient or improper staying, thus causing the plates to spring back and forth as the steam pressure varies. This is most commonly found in stationary boilers of the Cornish and Lancashire types. It appears on the flat end plates around the edge of the angle iron, or in the root of the angle iron. Too rigid staying of the ends by gusset or diagonal stays, or too great a difference in expansion of the tubes and shell is almost sure to cause grooving. Grooving may be due either to excess or to lack of stiffness. Grooving often occurs at the transverse joints when the riveting has not been properly done. Internal grooving is a direct result of excessive calking, which by injuring the surface of the metal, exposes it to the corrosive action of the feed water. When the feed water causes a thick layer of scale, if unequal expansion cracks it and exposes a fresh surface to corrosive action, grooving is likely to appear.

Sometimes grooving is so fine as to appear to be a crack or fracture. If the crack is only $\frac{1}{64}$ of an inch wide, it may extend into the plate for a considerable depth. On account of the minuteness of the cracks, they are likely to escape detection and when once formed, rapidly enlarge from any acidity in the feed water.

The best way to prevent internal corrosion is to use water which has no corrosive effect on the plates. If internal corrosion has begun, a change of water will often prolong the life of the boiler. In some instances it is cheaper to build a new boiler every few years than to use a different water supply. This is not always best, for there is a tendency to use the boiler as long as possible, even after it has become unsafe. Sometimes the introduction of a thicker plate at places where the water is found to

effect them will be economical; but as these plates will be stronger the strain will not be uniform. Another method is to neutralize the acidity of the feed water by some alkaline substance. This is done by dissolving soda or soda-ash in the feed water before it enters the boiler. The amount of soda to be used varies with the amount of acid. If there is considerable salt in the water, the introduction of soda is injurious.

Incrustation. The hard scale formed by the accumulation of the deposit of sediment from the feed water is sometimes called **scurf, fur,** or **incrustation.** The solid matter of the feed water is precipitated by the rise of temperature, or is left behind by evaporation. These solids, unless blown out, become hardened and form scale.

The amount of solid matter in solution is measured in **grains per gallon;** usually 20 to 40 grains per gallon. The quantity varies considerably. It is not the amount but the nature of the solids that determines the fitness of feed water. With proper attention to blowing off, the presence of a certain amount of carbonate, or chloride of soda would not be injurious, while the same number of grains per gallon of salts of lime would cause much trouble. **Salts of lime** are the ones most frequently found. The chief troublesome impurities found in feed waters for land boilers are sulphate and carbonate of lime, carbonate of magnesia, and some organic impurities. Marble and limestone are other names for carbonate of lime.

Water may be divided into two classes; **hard** and **soft.** Hard waters contain considerable lime, while soft waters have but little solid matter in solution.

The **carbonates of lime** and magnesia are held in solution in water by an excess of carbon dioxide. Heating the water will drive off the excess of carbon dioxide, or carbonic acid, and most of the carbonates will be precipitated. The precipitate is a whitish or grayish mud. As the temperature **increases,** the solubility **decreases,** and at the boiling point it is scarcely soluble. If the carbonates are not mixed with impurities, they may be washed out of the boiler after it has been allowed to cool. But if there is oil, organic matter, or sulphate of lime with them, the deposits are likely to become hard.

Sulphate of lime, like the carbonate is precipitated when the temperature rises. If water containing sulphate of lime, which forms a hard adhering scale, is heated to 280°F. all the lime will be precipitated. If a little carbonate of soda or soda-ash is introduced with the feed water, calcium carbonate is precipitated in the form of a fine white powder which may be washed out. The amount of carbonate of soda to be used may be calculated if the mineral impurities of the water are known. The soda should be introduced at regular intervals. The measured or weighed portion is dissolved in the water and then pumped into the boiler. An excess of soda is likely to cause foaming, and often sticks in the water glass. These indications show too much soda, but do not indicate when the right amount is being used.

Sulphate of lime forms an amorphous crust, or scale. The hardness depends upon the other impurities and the temperature. If water that contains lime, forms a hard scale, sulphates are present, and the amount of soda to be used may be determined by trial. The deposits of the carbonate of lime are soft and granular, and of grayish color. Sulphate of lime forms a hard crust. Ammonium chloride, commonly called sal-ammoniac, is sometimes used to break up the lime compounds but is not always desirable as it does not act on the sulphate and may break up the chlorides, thus forming free chlorine which attacks the boiler.

Carbonate of magnesia is seldom found in as large quantities as the salts of lime. It is like carbonate of lime in solubility.

When the feed water contains iron salts the incrustation is of a reddish color. Water containing these salts is generally very injurious to the boiler plates. Brackish water containing chloride of magnesia is also injurious, for when heated, the chloride decomposes, forming magnesia and hydrochloric acid which rapidly corrodes iron.

A piece of thick scale broken from the plates of a boiler, shows a series of layers of varying thicknesses. Some of them may be crystalline and some amorphous. Between these hard layers are sometimes found layers of soft or earthy matter.

The oil used in the engine is likely to get into the boiler if a surface condenser is used. If the feed water has a little organic matter or lime salts in it, the effect of the oil is increased. The

deposit of oil mixed with other impurities is a poor conductor of heat and the resultant overheating of the plate often causes bulging, as shown in Fig. 62.

A very thin coating of scale is generally regarded as benefi-

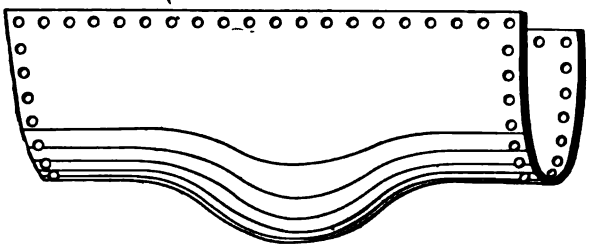


Fig 62.

cial rather than harmful, because it protects the plates from the corrosive action of the water. A thick scale however, reduces the conducting power of the plates, scale being a poor conductor of heat. A thick coating is not only uneconomical, but is dangerous since the heat from the furnace may burn the plates instead of being conducted to the water.

In regard to the loss of heat caused by scale on the heating surface, nothing is definitely known. There are too many circumstances to be considered to admit of exact calculation. It is stated that a layer $\frac{1}{16}$ of an inch on the tubes of multitubular boilers is equivalent to a loss of 15 to 20 per cent. of fuel, and the loss increases rapidly with the thickness. Also that when $\frac{1}{4}$ of an inch thick, 60 per cent. more fuel is required, and when $\frac{1}{2}$ inch thick, 150 per cent. The above statements indicate the loss by incrustation, but the figures cannot be relied upon, because neither the kind of boiler, nor the kind of incrustation is stated. Some boilers would work very well with $\frac{1}{2}$ an inch scale on the furnace plates while others would not work at all. Boilers have been worked with scale $\frac{1}{2}$ inch thick without a loss of even 100 per cent. of fuel. A uniform coating is not nearly as harmful as irregular deposits. The deposit that produces the effects of overheating where it is least suspected is a powder composed chiefly of carbonates of lime and magnesia. As it is very light, it does not sink to the plates, but is held in suspension or covers the surface as a scum. If this powder becomes mixed with oil or grease the

resistance offered to the rise of bubbles of steam is considerable. A layer of steam may thus be formed above the plates and the water lifted from them. As steam is a poor conductor of heat, the plates are likely to become overheated.

A compact homogeneous mass like hard scale is less harmful than a spongy powdery mass. This is readily seen from the fact that clean water can be boiled in a tea kettle having scale $\frac{1}{4}$ inch thick, with less danger of burning than can milk or water thickened with flour or oat-meal in a clean vessel. Loose sand is not as good a conductor of heat as solid rock.

Incrustation may be prevented, *1.* By precipitating the scale-forming substances before the feed water enters the boiler; *2.* By removing the sediment before it becomes hard scale, and *3.* By removing the hard scale from the interior of the boiler.

Water may be purified to a considerable extent by passing it through a purifier before allowing it to enter the boiler. The carbonate and sulphate are precipitated by heat. If the water contains much organic matter, it is advisable to filter it.

The precipitation of scale formation and the neutralization of acids in the feed water are accomplished to a greater or less extent by various chemicals. Soda, either as carbonate, soda-ash, or caustic soda is perhaps the most common. When soda and sulphate of lime are mixed, they exchange their acids and form sulphate of soda, which is soluble, and carbonate of lime which is precipitated. The action on the carbonate of lime is the precipitation of the lime salts. Caustic soda if concentrated may have a corrosive action. It should be used in smaller quantities than common soda. Potash, chloride of barium and tannic acid are also used, but soda is the most common. When tannic acid is used its action should be closely watched.

After the impurities are deposited in a boiler, the easiest way to remove them is by the **blow-out apparatus.** Impurities are held in suspension and float at the top of the water for considerable time before settling; the surface blow-out is used in this case.

It is sometimes stated that the loss by frequent blowing off is greater than that caused by scale. The loss of heat occasioned by blowing out a large quantity of water is considerable and whether or not it is advantageous to do so depends upon the particular case.

If lime and magnesia salts are to be blown out, the blow-cut taps should be opened frequently; about every hour.

A rapid **circulation** of the water prevents the formation of scale. The sediment is swept through the tubes and shell until it enters the mud drum or is blown off. This is one of the chief advantages claimed for water tube boilers.

The suspension of blocks of wood, sheet iron, zinc, etc., to catch the deposits is sometimes beneficial.

Zinc is used to a considerable extent to prevent corrosion and incrustation. When zinc is placed in connection with the plates, galvanic action is set up which destroys whatever action there may have been between the plates and fittings, or between the different portions of the plates. It is also claimed that the galvanic action prevents the formation of scale by liberating hydrogen at the exposed surfaces.

When incrustation is once formed there are several methods for removing it. It may be chipped off carefully with suitable tools: this operation is safe but laborious. This method is difficult if the boiler is complicated, or if the scale adheres closely. If the chipping is not carefully done, the surface of the plates and the rivet heads may be injured. If the surface of the iron is broken, the next formation of scale will cling more closely and any corrosive agent in the feed water will have a better opportunity for attack.

Another method of removing the scale is by sudden expansion and contraction of the plates. This is accomplished by blowing off, then admitting a large volume of cold water, or by cooling down as quickly as possible after the fire is drawn. These processes will generally remove the scale in large pieces, or loosen it so that it can be easily removed; but they are very harmful to the boiler and have led to destruction of boilers and loss of life.

The **simplest** method is to allow the boiler to cool as gradually as possible and then stand full of cold water for a few days The deposits are thus prevented from being baked hard on the plates and the sulphate of lime has an opportunity of re-dissolving in the cold water. On emptying the boiler the sulphate of lime is removed with the water and the deposits are washed or brushed out.

Kerosene oil is used to a considerable extent to prevent and

. emove scale. It breaks up and loosens hard scale and also prevents its formation. It is claimed that one quart per day for a 100 horse-power boiler is sufficient.

The following table gives a brief review of common corrosion, incrustation, causes and remedies:

TROUBLE.	CAUSE.	REMEDY.
Corrosion.	Chloride and Sulphate of Magnesium.	Carbonate of soda.
	Acids.	Alkalies.
	Grease.	Carbonate of soda. Filter. Use mineral oil.
	Carbonic acid and Oxygen	Heat feed water. Soda or slacked lime.
	Organic matter.	Precipitate with alum and filter.
Incrustation.	Sulphate of lime.	Carbonate of soda.
	Carbonates of lime, Magnesium and Iron.	Heat feed water. Soda, lime, or magnesia.
	Sediment, mud, etc.	Blow off, or filter.
	Any soluble salts.	Blow off.
		In general, kerosene, chipping, etc.
Priming.	Organic matter.	Precipitate with alum and filter.

Fractures. There are two classes of fractures. *1.* Those caused by lack of freedom for expansion and contraction. *2.* Those caused by inability of the material to bear the steam pressure.

In an internally fired boiler, the internal flue may expand considerably, while the shell, held by stays and the ends, will

expand but little. When the fire is external, *i. e.*, under one end, the plates just above the fire being very hot expand more than the rest of the shell or the tubes. If the tube plates cannot expand and contract, the tube holes are likely to become oval and leak. Furnace plates fracture from sudden contraction caused by cooling of the plates when the furnace door is opened.

Plates usually fracture from the holes to the edge of the plate in the outside lap. Fractures are not always dangerous, but do need repairing. Usually they are small and at first barely visible, but soon they enlarge and when increased by corrosion cause leakage and perhaps rupture. When the lap-edge fractures extend through the rivet holes into the plate, drilling holes and placing rivets in their path will usually stop them. When the fractures run in a line between the rivet holes, they are dangerous. A good plate will usually leak and give indication of danger, while a brittle plate gives way suddenly. Thick plates are more likely to fracture than thin plates.

Bad workmanship and brittle plates are common causes for fractures. Careful supervision and the employment of skilled workmen are the only remedies for this evil. In case a plate is punched at the wrong points and then punched over again it is greatly weakened. This defect is often concealed by an angle iron.

Sometimes a boiler will rupture immediately after repairs. The plates are often injured by the methods used to knock off the rivet heads and force out the old rivets. When the new plates are put on, the eckless use of the drift increases the injury.

BOILER EXPLOSIONS.

The first requisite of a steam boiler is **safety.** This condition is not always fulfilled, but the frequent disastrous explosions prove its necessity. When a boiler explodes, the first question asked is, "*What caused it?*" Answers differ widely, and sometimes the cause is unknown. Those interested are often unwilling that there should be an investigation, thinking the responsibility may be placed on them.

Since the explosion of a boiler in a city is likely to prove fatal to many people, and cause much destruction, the responsi-

bility of the boiler owner is not slight. The owner should be held liable in case of explosion, if he uses a boiler of poor type, a boiler which is not regularly inspected, a condemned boiler, or if he employs a man to take charge of it who holds no license.

The damage done by explosion is due to the energy stored in the hot water. The amount of energy in hot water is calculated by thermodynamic methods. The energy in one pound of hot water at 150 pounds pressure and 358°F., is about 10,500 foot pounds, *i. e.*, it is sufficient to raise one pound to a height of nearly two miles, and if at 250 pounds pressure, it has sufficient energy to raise it nearly three miles.

Energy. The energy stored in a Lancashire boiler 30 feet long and 8 feet in diameter, with 150 pounds pressure, has been calculated to be about 207,600 foot-tons, which is sufficient to raise the whole boiler 12,500 feet, or more than two miles. It takes 1,730 pounds of gun-powder to give an equivalent energy. The energy in a marine boiler 15 feet long, 13 feet in diameter and with a pressure of 150 pounds, is equal to 2,464 pounds of gun-powder. An ordinary return tubular boiler under 75 pounds pressure, has within it sufficient energy to blow it over a mile into the air. The above figures give some idea of the enormous amount of energy in a boiler, and the destruction caused by its explosion.

If a boiler filled with steam and hot water, fractures, or the pressure becomes more than the plates can withstand, a violent explosion generally follows. The steam escaping through the rent, or opening, diminishes the pressure slightly, thus causing a large quantity of steam to be generated from the hot water in a very short time. The instantaneous formation of a large quantity of steam produces disastrous results. The larger the body of hot water, the greater the disaster on account of the stored energy, sometimes called thermal storage. This fact accounts for the safety of water tube boilers; the division of the water into small masses in the sections prevents violent explosion. The bursting of one of these sections is unusual, but in case it happens the damage rarely extends beyond the boiler room; sometimes it merely causes a rush of steam into the room. This is disagreeable but not fatal. The explosion of a shell boiler usually wrecks the

whole building. A small part of the stored energy is used in smashing up the boiler; the larger part scatters the fragments.

Causes. The general cause for boiler explosions is over pressure of steam. This over pressure is due to the inability of the boiler to stand the working pressure, or to the failure of the safety valve to blow off at the required pressure.

The reasons why a boiler is unable to bear the working pressure are: *1.* Defective design, or ignorance as to the amount of pressure the boiler will stand. *2.* Deterioration, that is, reduction of strength from corrosion, incrustation, and wear. *3.* Defective workmanship or material. *4.* Mismanagement.

Defective Design includes the following faults. The boiler may be insufficiently or improperly stayed; the stays being too small or too few, or they may be bent or cut to clear floats, pipes, etc. Flat surfaces may be stiffened but not stayed. The internal furnace flues may not have sufficient strengthening rings. The large holes for manholes, domes, etc., may be cut in the plates without the edges being properly strengthened. The boiler may be set too rigidly and no provisions made for expansion and contraction of the plates. Defective water circulation causes overheating, excessive incrustation, etc. The feed apparatus, not being in good working order, causes low water, overheating, rupture, and explosion. A badly designed safety valve may cause considerable trouble if not disaster.

Deterioration. The strength of a boiler may be impaired by fractures, general corrosion, pitting, or grooving. External corrosion especially, is the cause of many disasters. It can proceed unseen in many cases and rupture may occur when least expected. In the investigation of boiler explosions, the plates have often been found much reduced in thickness. When incrustation and grease have formed a pocket as previously explained, neglect to repair will usually result in rupture or explosion. The admission of cold feed water on hot plates is likely to cause fractures, which enlarge by fluctuation of pressure and corrosion until the boiler becomes too weak to bear the steam pressure.

Vertical boilers are the most likely to explode. The upper ends of the tubes, if they pass through the steam space, by collapsing, will cause disaster. The crown sheet is likely to corrode

and bulge downward; the reaction of the escaping steam may throw the boiler high in the air.

The strength of a boiler, originally sufficient to stand high pressure, may become suddenly reduced by over-heating or over-straining. *Over-heating* makes the plates soft and weak. It may be caused by lack of water, accumulation of sediment or scale on the furnace plates or flues, defective circulation, or the concentration of heat on thick plates. *Over-straining* is caused by cooling and contraction, or by sudden expansion of tubes or flues.

Shortness of water, or low water, is the result of leakage of joints, valves, taps, etc., or of neglect to close the valves. It is also caused by sudden demand for steam, failure of the feed supply or check valve, the water siphoned from one boiler into another, defective gages, etc.

Defects of workmanship and material are usually found in flue boilers. They can be readily examined inside, and also subjected to the hydraulic test. Defects of workmanship are: careless punching and riveting, fractured rivet holes, broken rivets, carelessly secured stays, fractures in flanging, bending, welding, hammering, etc. Patching old plates with new ones, seriously damages them, especially if the removal of the old rivets and the placing of the new ones is not carefully done. The greater contraction and expansion of the new plate causes a severe local strain.

Defects of material are not easily discovered. Ordinary inspection will not reveal brittleness. Blisters, laminations, and adhesion of foreign substances during manufacture are usually detected during inspection.

Mismanagement causes many explosions. The pressure in a steam boiler may rise above the blowing off pressure by rusting of the safety valve, by poor design, by intentional or accidental overloading of the safety valve, by placing it on the steam pipe, or by placing a stop valve between the boiler and the safety valve. In one instance a safety valve, recovered from an explosion was corroded to such an extent that 3,000 pounds pressure was required to start it from its seat. When the size or lift of the safety valve is small, steam may be generated faster than it can escape, and cause a rapid increase of pressure. The lever safety valve is easily

overloaded, either by an increase in length or by an addition of weights. It also may be wedged down. The safety valve is sometimes overloaded to make the blow off pressure equal the pressure indicated by a defective pressure gage. It is often loaded or wedged down to prevent leakage at the working pressure, in case it needs regrinding. The sudden opening or closing of a large steam or safety valve may convert static pressure into dynamic force, which may produce a violent rush of steam and hot water against some part of the boiler. This pressure may be enormous, causing rupture and perhaps explosion.

The practice of placing a stop valve between the boiler and the safety valve cannot be condemned too strongly. During repairs the stop valve may be shut and the communication temporarily shut off. Neglect to open the valve on resuming steam pressure is likely to cause serious trouble.

It is a common idea that when the furnace plates become very hot, heated to redness for instance, and the feed turned on, a violent explosion is sure to follow. Experiment shows that when a piece of wrought iron is heated to redness and plunged into a weight of cold water, three or four times greater than that of the iron, a very small quantity of steam is disengaged. Why should the quantity of steam be greater if the iron is in the form of a boiler? If a boiler contains considerable hot water, heated to 400°F. for instance, the addition of cold water might be followed by an explosion. If the body of water admitted to hot plates is small, there is greater danger of disaster.

The following experiments illustrate the action of cold water on hot plates. A boiler 25 feet long and 6 feet in diameter was heated red hot and the feed turned on. No explosion occurred; but the sudden contraction of the overheated plates caused the water to pour out in streams at every seam and rivet hole as far as the fire mark extended. In another instance, the water was almost entirely drawn off while the fires were burning briskly. When the remaining water had been converted into steam and all the fusible plugs melted out, water at the rate of 28 gallons per minute in a series of fine jets was played on the hot plates. The only effect was to reduce the pressure.

That a tough paper or cloth is easily torn when once started

is a well-known fact. Similarly a boiler plate may be ruptured at slight pressure if the fracture has been started.

The **position of the fracture** or hole has a great influence on the results. In case a large rent occurs at the top of a cylindrical boiler, the steam and hot water may blow out of the hole, and the boiler, if strongly enough seated to stand the recoil will remain on its seat. The damage to the boiler would be slight. But suppose the same rent were situated on the under side of the boiler near the ground or floor; the effect would be very different, the reaction of the escaping steam would probably blow the whole boiler through the roof.

Investigation. When an explosion occurs it should be **investigated**; not only to fix the responsibility where it belongs, but also to provide for and take means to prevent future disasters. It has been customary to attribute all explosions to low water, since it is an easy way to throw the responsibility from the makers or owners upon the fireman, who, even if living, cannot defend himself. In the investigation of an explosion the weights, shapes, positions, and directions of the scattered pieces should be noted, so that their original places may be known. The original size and shape of the boiler and of the fittings should be known as accurately as possible. The primary rent may be discovered from comparison and from deductions of the directions taken by the heavier pieces. Light pieces will generally take the direction of the escaping steam, while the heavy parts take an opposite direction, that of the reaction. A careful examination of the pieces, noting the age of fractures, thickness of plates, amount of corrosion, condition of plates, etc., will generally show the cause. A test of the plates will in many cases show any softening or yielding to the pressure and excessive thinness caused by bulging.

Prevention. The means taken to prevent boiler explosions from most of the above mentioned causes have been already given. It is of primary importance that at the start only a well-designed and well-made boiler should be used. The matter of type is not of so much importance, but it is well to use a sectional boiler in large cities or in buildings where many people are employed. There are many methods, some of which have been discussed, that

are taken to prevent deterioration by corrosion, fracture, etc. Proper setting is of great importance in this matter. Mishaps from mismanagement may be greatly lessened by the employment of licensed attendants. A boiler should *never* be in the hands of a man who is not thoroughly competent to run it. The most effective method to prevent explosions is the law of the state, compelling regular, thorough inspection and licensed firemen. The inspection by the Boiler Insurance companies is also an efficient method.

During a period of eleven and one-half years 70,000 boilers were inspected by Boiler Insurance companies. It was estimated that there were 140,000 in use during that time. Of the inspected boilers, there were 23 explosions and 50 collapses; resulting in 27 deaths from explosions, and 28 deaths from collapses. The explosion rate was 1 in 11,000 and the death rate 1 in 14,600. The uninsured boilers did not make as good showing; the death rate being 1 in 5,000 boilers, or about 3 times as high as among the insured boilers.

FUELS.

Coal is the most important fuel and in many localities the *only* one. As the population increases and the demand for timber increases, the use of wood as fuel for boiler furnaces diminishes. In some places wood, gases, and oils are abundant, and therefore most used. For some manufacturing plants, the waste products, such as straw, tan, sawdust and bagasse are the most economical.

There are several kinds of coal, all of vegetable origin, but differing according to the geological changes in the locality in which they are found.

There are five chief kinds of coal;

1. **Anthracite**, a hard variety, consisting almost entirely of free carbon.
2. **Dry bituminous**, having 70 to 80 per cent. of carbon.
3. **Caking bituminous**, having 50 to 60 per cent. of carbon.
4. **Cannel coal**, having 70 to 85 per cent. of carbon.
5. **Lignite**, or brown coal, containing 56 to 76 per cent. of carbon.

Anthracite Coal consists almost entirely of carbon and has but little hydrocarbon. Good anthracite is hard, lustrous, com-

pact, and breaks up easily under a high temperature. It burns with very little flame and smoke, unless wet, and gives an intense heat. On account of its brittleness, the fine pieces will be lost unless the spaces in the grate are small, or a down draft furnace having double grates, is used.

A variety of coal called semi-anthracite or semi-bituminous, which contains some hydrocarbon, is not as compact as good anthracite and burns with a short flame. Anthracite coals are found chiefly in the Allegheny Mountains and in the Rocky Mountains of Colorado.

Bituminous Coal contains a large and varying amount of hydrocarbon, or bituminous matter.

Dry bituminous coal burns freely, with little smoke and without caking. It is found in Maryland and Virginia.

Caking bituminous becomes pasty, swells up and cakes when burning. It is used for making illuminating gas, and is mined principally in the Mississippi Valley.

Cannel Coal, or long flaming bituminous, has a tendency to produce smoke; some varieties cake. Cannel coals are found in Pennsylvania, Indiana, and Missouri.

Lignite, or brown coal, is intermediate between coal and peat. It is made up mostly of carbon, with some moisture and mineral matter. Poor varieties are of little value. Lignites come from Colorado, Texas, and Washington.

There are a few solid fuels which are of importance, although they are but little used as a boiler fuel.

Coke is made by driving off, by heat, the hydrocarbon of bituminous and semi-bituminous coals. If made in gas retorts, it is weak, and is but little used for making steam. When made in coking ovens, it is dark gray, hard, porous, and brittle. It gives out a slight ringing sound when struck, and has a metallic lustre. If the coal is moist, or if steam is used in coking, as in the manufacture of water gas, the sulphur is burned out. Coke burns without flame, and will, if forced, make a hot fire.

Charcoal. Charcoal is seldom used in the steam boiler furnace. It is made by evaporating the volatile constituents of wood, either by partial combustion or by heating in retorts. If it is made in conical heaps, 100 parts of wood by weight, yield about

20 parts of charcoal; if made in retorts about 30 parts of charcoal are realized. About 50 bushels of charcoal are obtained from a cord of wood.

Peat. Peat, or turf, consists of decayed roots and earthy matter. It is found in bogs and swamps. After it is cut out and dried, it is sometimes pulverized and compressed as an artificial fuel. The amount of ash is from 5 to 11 per cent. Its specific gravity is from .4 to .5, but it can be compressed to greater density.

Wood. There are two principal divisions of wood. Hard wood, which is compact and comparatively heavy, as oak, ash, etc., and soft wood, which is light colored, soft, and of less specific gravity than hard woods. Pine, birch, and poplar are examples of soft woods. Wood contains considerable moisture, even if left in a dry place for a few years. After being thoroughly dried, it will absorb and retain from ten to twenty per cent. of moisture. The chemical composition of different woods is nearly alike and is about as follows:

	Perfectly Dry Wood.	Ordinary Fire Wood.
Carbon	50 per cent.	37.50 per cent.
Hydrogen	6 "	4.50 "
Oxygen	41 "	30.75 "
Nitrogen	1 "	.75 "
Ash	2 "	1.50 "
Moisture	. . . . .	25.00 "

English oak, a hard wood, has a specific gravity of .93 and weighs 58 pounds per solid cubic foot. Yellow pine, a soft wood, weighs 41 pounds per solid cubic foot. Its specific gravity is .66.

A cord of wood measures 4 feet by 4 feet by 8 feet and equals 128 cubic feet. For ordinary pine wood, one cord weighs 2,700 pounds, or 21 pounds per cubic foot.

The heating value for a given weight of different kinds of wood is very nearly the same. That is, one pound of hickory is worth no more as fuel than one pound of pine, if both are dry. Wood that is kiln dried has a theoretical heat value of almost 8,000 B. T. U., while wood containing about 25 per cent. moisture has a value of about 6,000 B. T. U.

The most important elements in coal are hydrogen, producing 62,000 B. T. U. per pound, and carbon, producing 14,500 B. T. U.

per pound. Although several coals may have the same total percentage of combustible and ash, the heat values may not be the same, because heat value depends upon the amounts of hydrogen and carbon they contain. The heat value of fuels is determined by chemical analysis, or by calorimetric test, and varies for coal from different localities. The following table is compiled from several sources.

Kind of Coal.	Per cent. of ash.	Theoretical B. T. U. per pound.	Pounds of water evaporated per pound. (Theoretical.)
Penn. Anthracite	3.49	14,199	14.70
Penn. Anthracite	2.90	14,221	14.72
Penn. Cannel	15.02	13,143	13.60
Penn. Connellsville	6.50	13,368	13.84
Penn. Semi-bituminous	10.70	13,155	13.62
Penn. Brown	9.50	12,324	12.75
Kentucky Caking	2.75	14,391	14.89
Kentucky Cannel	2.00	15,198	16.76
Kentucky Lignite	7.00	9,326	9.65
Indiana Caking	5.66	14,146	14.64
Indiana Cannel	6.00	13,097	13.56
Maryland Cumberland	13.88	12,226	12.65
Arkansas Lignite	5.00	9,215	9.54
Colorado Lignite	9.25	13,562	14.04
Texas Lignite	4.50	12,962	13.41
Washington Lignite	3.40	11,551	11.96

In practice, no fuel gives its theoretical evaporation value on account of several losses. Heat is radiated from, and conducted away by the boiler setting. The admission of air into the furnace, either through the doors or through cracks in the setting, reduces the theoretical evaporation value. Improper firing, either by carelessness or ignorance, causes considerable loss, and errors in design, construction or of setting reduce the efficiency. The loss from radiation has often been found to be 20 per cent. A boiler seldom gives an efficiency of 80 per cent., the average being 65 to 70 per cent.

Tan. Bark after it has been used in the process of tanning leather is called tan. This tan which is the fibrous part of the bark, is sometimes burned in the boiler furnace. The heat-

ing power of dry tan, containing 15 per cent. of ash, is about 6,000 B. T. U. If the tan is in the ordinary state of dryness, that is, containing 25 to 30 per cent. of moisture, the heat value is about 4,300 B. T. U. Perfectly dry tan will evaporate about 6.3 pounds of water supplied at 212°F. If the tan contains 30 per cent. of moisture, it will evaporate only 4.4 pounds.

Straw. Straw, in its ordinary condition, is composed as follows:

Carbon	36.	per cent.
Hydrogen	5.	"
Oxygen	38.	"
Nitrogen	.425	"
Ash	4.75	"
Water	15.75	"

The heat value for ordinary straw is about 8,000 B. T. U., equivalent to the evaporation of about 7.3 pounds of water from and at 212°F.

Bagasse is the fibrous portion of the sugar cane left after the juice has been extracted. It usually contains 25 to 40 per cent. of water. On account of the large amount of water, it must be dried before being used in the furnace. Bagasse is equal in calorific value to about one fifth the same weight of coal. This value varies with the chemical composition. The larger the per cent. of extraction of the sugar, the drier the bagasse and the greater its evaporative power. It is therefore of great advantage to extract as much of the juice as possible.

Cook's automatic apparatus burns the bagasse direct from the sugar mill. This saves the labor of oxen and men required for gathering, spreading and firing. The amount of sugar, if the extraction is not complete, is sufficient, when used as a fuel, to evaporate the moisture. The combustion by this method is better than that obtained from any hand firing. The waste heat that escapes to the chimney is utilized in heating the air admitted to the furnace. This hot blast is very efficient when burning wet fuel.

Bagasse should be fed continuously into a very hot fire-brick chamber which has plenty of room to accommodate the large volume of gas and steam produced by the combustion.

Petroleum. Oil is now becoming a fuel of considerable importance. Petroleum is a hydrocarbon liquid composed of carbon, hydrogen and oxygen in slightly varying proportions.

The following is the composition of Petroleum.

Carbon	82 to 87 per cent.
Hydrogen	11 to 15 per cent.
Oxygen	.5 to 6 per cent.

The theoretical heating power is about 20,000 B. T. U. per pound, which is equivalent to the evaporation of 21 pounds of water from and at 212°F.

Petroleum oils are obtained from the distillation of petroleum and are compounds of carbon and hydrogen. The compounds varying from C_{12} H_{24} to C_{32} H_{64}; or expressed in weight,

Carbon	from 71.42 to 73.77 per cent.
Hydrogen	from 26.23 to 28.58 per cent.

The specific gravity of petroleum is about .87 and that of the petroleum oils from .628 to .792. The heating value of petroleum oils is from 27,000 to 28,000 B. T. U., and the equivalent evaporation 24 to 25 pounds when water is supplied at 62°F., or 28 to 29 pounds when the water is at 212°F.

There are several advantages claimed for oil fuels, but there are also some disadvantages.

Advantages. A petroleum fire can be controlled by turning one valve. If the fire is regulated to produce the required heat, it can be kept at that point with very little care. The fire can be started with slight trouble and, if not wanted continuously, can be extinguished instantly. As in the case of the gas engine, there is no waste of fuel after the work is done.

Petroleum produces a uniform fire, much more so than that of coal or wood.

Coal contains about 14,300 heat units, of which about 8,000 are available, the rest being lost in slate, sulphur and earthy matter. Petroleum, it is claimed, furnishes 20,000 B. T. U. and is equivalent to more than two pounds of coal.

A pound of *coal* in an ordinary boiler will evaporate from 7 to 11 pounds of water, while *oil tests* show 17 or 18 pounds.

Complete combustion, cleanliness and economy of labor are obtained by the use of petroleum.

The heat in the furnace does not fluctuate since there is no need of opening doors. The burning of petroleum is not as harmful to the furnace as the burning of coal, because the oil has less sulphur. The saving from handling is an important item, especially noticeable in marine work.

Among the **disadvantages** are: Danger from explosion on account of the vapors which rise from the fuel; loss of fuel by evaporation; unpleasant odor; comparatively high price.

The question of price of fuel is dependent upon locality, and must be determined for each case.

Mineral oil as crude petroleum, or the refuse heavy oil left from the distillation of petroleum oils is sometimes used as boiler fuel. The oil is thrown into the furnace as finely divided spray by means of special apparatus, in which superheated steam or compressed air is the motive force. Superheated steam is more convenient, but it is not as good on account of the inert material (moisture) to be heated.

Gas. Gas has many advantages over any other kind of fuel. Some think it an ideal fuel. Natural gas has been found in Pittsburg, Pennsylvania, and Indiana, in such quantities that many manufacturing industries use it exclusively. There have been many processes invented to produce artificial gas that would be cheap enough to compete with coal. Coal gas, made by the distillation of coal, and water gas, obtained by the decomposition of steam by incandescent carbon, have been used both for lighting and for fuel. In the following table of relative heat values, the water gas is not carbureted. When carbureted, its value may exceed that of coal gas. Producer gas is made by blowing steam and air through the coal.

	By weight.	By volume.
Natural gas	1000	1000
Coal gas	949	666
Water gas	292	292
Producer gas	76.5	130

The natural gas in the preceding table is from Findlay, O. The coal gas is an average sample used as an illuminating gas. The water gas is made for heating and not purified for illuminating. The producer gas is an average sample from the Pennsylvania Steel works, made from anthracite coal.

The value of natural gas is due to the large amount of marsh gas it contains, since the marsh gas has a large per cent. of hydrogen. On account of its large per cent. of hydrogen, coal gas is nearly equivalent to natural gas.

The relative values of these gases for evaporation are shown in the following table:

	Natural Gas.	Coal Gas.	Water Gas.	Producer Gas.
Cubic feet of gas	1000	1000	1000	1000
Pounds of water evaporation	893	591	262	115

Experiments in Pittsburg show that 1000 cubic feet of natural gas equals 80 to 133 pounds of coal. The coal used in the comparison varied from 12,000 to 13,000 B. T. U. per pound.

It has been stated by the Western Society of Engineers that one pound of good coal is equal to 7½ cubic feet of natural gas.

As in the case of petroleum, the economy of burning gaseous fuels depends upon the locality.

Artificial fuels. The waste of charcoal, coal, sawdust, etc., is often pressed into cakes, or briquettes, by means of some adhesive mixture with compression. Wood tar, coal tar and clay are used according to convenience. These cakes are compact, can be stored in a small space, and are used where good fuels are difficult to obtain.

TESTING STEAM BOILERS.

A steam boiler may be subjected to two tests, namely; the **strength**, and the **efficiency** test.

Even if the boiler has been carefully designed and the workmanship and every detail of construction carefully inspected, the safety can only be ascertained by the application of pressure. If every piece of metal has been tested and the parts put together according to best results of long experience, there may, however, be defects, such as bad welding, careless riveting, or plates cracked in flanging, which will greatly impair the strength of the boiler.

There are certain classes of boilers so constructed that it is almost impossible to inspect them from within; of which the locomotive, crane, and small agricultural boilers are examples.

Many hidden defects are brought to light by careful inspection and the pressure test.

There are two ways of applying the pressure test, by *steam* and by *water*. In testing by steam, the conditions of strain are the same as those under which the boiler is worked. But the practice of testing a boiler with twice the steam pressure that is required, is likely to weaken if not explode it.

One method of testing is to pump water into the boiler until the desired pressure is reached. The distortions or defects of joints are then noticed. In testing a new boiler, the various parts should be carefully measured and gaged, and the results recorded. After the test has been maintained for some time, the measurements should be taken again and compared with the original. Any extension or bulging should be noted. On release of pressure, the measurements should be taken a third time to find the permanent set of any part. In case of permanent set, the material should be carefully tested, to see whether the permanent set was due to defects of construction, or to an exceeded elastic limit. There are instances, stayed surfaces for example, when a slight permanent set does not render the boiler unfit for use.

If the boiler leaks under the hydraulic pressure, the leaky joints should be marked and calked when the pressure is off.

Another method is to fill the boiler with water, close every outlet and then build a fire in the furnace. Water expands $\frac{1}{24}$ of its volume when the temperature rises from 60° F. to 212° F. When the water in the boiler is heated there is a corresponding increase of internal pressure, and the condition of the boiler is judged from the movement of the pointer of the pressure gage. Steady rise of pressure indicates soundness, while jerks or starts of the pointer are considered to denote weakness. This method is seldom used because of the difficulty of regulating the fire so as to give a steady increase of heat and of keeping that heat constant for any length of time. Although no explosion can result from hydraulic tests, a serious accident can happen when the plates fracture. Many say that the hydraulic test is worthless because it does not show the actual strength. The most injurious stresses, those due to unequal expansion, are not set up during this test. In order to find out the exact strength, the boiler must be destroyed, or at least so fractured or strained as to become worthless.

The application of high pressure is likely to strain the boiler and to render it unfit for the intended low pressure.

A boiler that has been at work for some time is considered safe if it will stand the test of a pressure one and a half times the working pressure.

The inspection of a boiler should commence at the works where the plates are manufactured. If the material and each detail of construction is inspected and tested, the boiler is likely to be strong.

The other test; the efficiency test, is of importance to boiler owners in order that they may know the boiler from the economic standpoint.

STEAM BOILER TRIALS.

Object. The object of a boiler trial is to determine the following; the *quantity* and *quality* of steam it will supply under given conditions; the *horse-power* of the boiler and its *efficiency*; and the *amount of fuel* it takes to make the required steam.

The quantity of steam is taken as the amount of water evaporated. The quality is determined by some form of calorimeter, already described. The efficiency is the ratio of the heat units utilized in evaporating the water, to the total heat supplied to the boiler. The heat utilized in evaporation is calculated by multiplying the number of pounds of feed water by the number of heat units required to change water, at the temperature of the feed water into steam at the gage pressure. The heat units supplied are determined by weighing the fuel used during the test, and after deducting the amount of ash, multiplying the result by the total heat of combustion of the fuel. The heat of combustion is obtained either by calculation or by means of a fuel calorimeter.

Great care and skill must be exercised throughout the test, for slight errors in temperatures, pressures, weighings, etc., will make the results worthless. It is much better to use a throttling or separating calorimeter, because the inaccuracy of the barrel calorimeter is likely to make a large error in the final results.

The Committee of the American Society of Mechanical Engineers recommended the following code of rules for boiler trials.

PRELIMINARIES TO A TEST.

1. In preparing for and conducting trials of steam-boilers the specific object of the proposed trial should be clearly defined and steadily kept in view.

2. Measure and record the dimensions, position, etc., of grate and heating surfaces, flues, and chimneys, proportion of air-space in the grate-surface, kind of draught, natural or forced.

3. Put the boiler in good condition. Have heating surface clean inside and out, grate-bars and sides of furnace free from clinkers, dust, and ashes removed from back connections, leaks in masonry stopped, and all obstructions to draught removed. See that the damper will open to full extent, and that it may be closed when desired. Test for leaks in masonry by firing a little smoky fuel and immediately closing damper. The smoke will then escape through the leaks.

4. Have an understanding with the parties in whose interest the test is to be made as to the character of the coal to be used. The coal must be dry, or if wet, a sample must be dried carefully and a determination of the amount of moisture in the coal made, and the calculation of the results of the test corrected accordingly. Wherever possible, the test should be made with standard coal of a known quality. For that portion of the country east of the Alleghany Mountains good Anthracite egg coal or Cumberland semi-bituminous coal may be taken as the standard for making tests. West of the Alleghany Mountains and east of the Missouri River, Pittsburg lump coal may be used.

5. In all important tests a sample of coal should be selected for chemical analysis.

6. Establish the correctness of all apparatus used in the test for weighing and measuring. These are: *1.* Scales for weighing coal, ashes, and water. *2.* Tanks, or water-meters for measuring water. Water-meters, as a rule, should only be used as a check on other measurements. For accurate work the water should be weighed or measured in a tank. *3.* Thermometers and pyrometers for taking temperatures of air, steam, feed-water, waste gases, etc. *4.* Pressure-gauges, draft-gauges, etc.

7. Before beginning a test, the boiler and chimney should be thoroughly heated to their usual working temperature. If the boiler is new, it should be in continuous use at least a week before testing, so as to dry the mortar thoroughly and heat the walls.

8. Before beginning a test, the boiler and connections should be free from leaks, and all water connections, including blow and

extra feed-pipes, should be disconnected or stopped with blank flanges, except the particular pipe through which water is to be fed to the boiler during the trial. In locations where the reliability of the power is so important that an extra feed-pipe must be kept in position, and in general when for any other reason, water-pipes other than the feed-pipes cannot be disconnected, such pipes may be drilled so as to leave openings in their lower sides, which should be kept open throughout the test as a means of detecting leaks, or accidental or unauthorized opening of valves. During the test the blow-off pipe should remain exposed.

If an injector is used it must receive steam directly from the boiler being tested, and not from a steam-pipe or from any other boiler.

See that the steam pipe is so arranged that water of condensation cannot run back into the boiler. If the steam pipe has such an inclination that the water of condensation from any portion of the steam-pipe system may run back into the boiler, it must be trapped so as to prevent this water getting into the boiler without being measured.

STARTING AND STOPPING A TEST.

A test should last at least ten hours of continuous running, and twenty-four hours whenever practicable. The conditions of the boiler and furnace in all respects should be, as nearly as possible, the same at the end as at the beginning of the test. The steam pressure should be the same, the water-level the same, the fire upon the grates should be the same in quantity and condition, and the walls, flues, etc., should be of the same temperature. To secure as near an approximation to exact uniformity as possible in conditions of the fire and in temperatures of the walls and flues, the following method of starting and stopping a test should be adopted.

10. **Standard Method.** Steam being raised to the working pressure, remove rapidly all the fire from the grate, close the damper, clean the ash-pit, and as quickly as possible start a new fire with weighed wood and coal, noting the time of starting the test and the height of the water-level while the water is in a quiescent state, just before lighting the fire.

At the end of the test remove the whole fire, clean the grates and ash-pit, and note the water-level when the water is in a quiescent state; record the time of hauling the fire as the end of the test. The water-level should be as nearly as possible the same as at the beginning of the test. If it is not the same, a correction should be made by computation, and not by operating pump after test is completed. It will generally be necessary to regulate the discharge of steam from the boiler tested by means of the stop-valve for a time while fires are being hauled at the beginning and at the end of the test, in order to keep the steam-pressure in the boiler at those times up to the average during the test.

11. **Alternate Method.** Instead of the standard Method above described, the following may be employed where local conditions render it necessary.

At the regular time for slicing and cleaning fires have them burned rather low, as is usual before cleaning, and then thoroughly cleaned; note the amount of coal left on the grate as nearly as it can be estimated; note the pressure of steam and the height of the water-level—which should be at the medium height to be carried throughout the test—at the same time; and note this time as the time of starting the test. Fresh coal, which has been weighed, should now be fired. The ash-pits should be thoroughly cleaned at once after starting. Before the end of the test the fires should be burned low, just as before the start, and the fires cleaned in such a manner as to leave the same amount of fire, and in the same condition, on the grates as at the start. The water-level and steam-pressure should be brought to the same point as at the start, and the time of the ending of the test should be noted just before fresh coal is fired.

DURING THE TEST.

12. **Keep the Conditions Uniform.** The boiler should be run continuously, without stopping for meal-times or for rise or fall of pressure of steam due to change of demand for steam. The draught being adjusted to the rate of evaporation or combustion desired before the test is begun, should be retained constant during the test by means of the damper.

If the boiler is not connected to the same steam-pipe with

other boilers, an extra outlet for steam with valve in same should be provided, so that in case the pressure should rise to that at which the safety-valve is set, it may be reduced to the desired point by opening the extra outlet, without checking the fires.

If the boiler is connected to a main steam-pipe with other boilers, the safety-valve on the boiler being tested should be set a few pounds higher than those of the other boilers, so that in case of a rise in pressure the other boilers may blow off, and the pressure be reduced by closing their dampers, allowing the damper of the boiler being tested to remain open, and firing as usual.

All the conditions should be kept as nearly uniform as possible, such as force of draft, pressure of steam, and height of water. The time of cleaning the fires will depend upon the character of the fuel, the rapidity of combustion, and the kind of grates. When very good coal is used, and the combustion not too rapid, a ten-hour test may be run without any cleaning of the grates, other than just before the beginning and just before the end of the test. But in case the grates have to be cleaned during the test, the intervals between one cleaning and another should be uniform.

13. **Keeping the Records.** The coal should be weighed and delivered to the fireman in equal portions, each sufficient for about one hour's run, and a fresh portion should not be delivered until the previous one has all been fired. The time required to consume each portion should be noted, the time being recorded at the instant of firing the first of each new portion. It is desirable that at the same time the amount of water fed into the boiler should be accurately noted and recorded, including the height of the water in the boiler and the average pressure of steam and temperature of feed during the time. By thus recording the amount of water evaporated by successive portions of coal, the record of the test may be divided into several divisions, if desired, at the end of the test, to discover the degree of uniformity of combustion, evaporation, and economy at different stages of the test.

14. **Priming Tests.** In all tests in which accuracy of results is important, calorimeter tests should be made of the percentage of moisture in the steam, or of the degree of superheating. At least

ten such tests should be made during the trial of the boiler, or so many as to reduce the probable average error to less than one per cent., and the final records of the boiler test corrected according to the average results of the calorimeter tests.

On account of the difficulty of securing accuracy in these tests, the greatest care should be taken in the measurements of weights and temperatures. The thermometers should be accurate within a tenth of a degree, and the scales on which the water is weighed to within one hundredth of a pound.

Analysis of Gases.—Measurement of Air Supply, etc.

15. In tests for purposes of scientific research, in which the determination of all the variables entering into the test is desired, certain observations should be made which are in general not necessary in tests for commercial purposes. These are the measurement of the air-supply, the determination of its contained moisture, the measurement and analysis of the flue gases, the determination of the amount of heat lost by radiation, of the amount of infiltration of air through the setting, the direct determination by calorimeter experiments of the absolute heating value of the fuel, and (by condensation of all the steam made by the boiler) of the total heat imparted to the water.

The analysis of the flue-gases is an especially valuable method of determining the relative value of different methods of firing, or of different kinds of furnaces. In making these analyses great care should be taken to procure average samples—since the composition is likely to vary at different points of the flue, and the analyses should be intrusted only to a thoroughly competent chemist, who is provided with complete and accurate apparatus.

As the determinations of the other variables mentioned above are not likely to be undertaken except by engineers of high scientific attainments, and as apparatus for making them is likely to be improved in the course of scientific research, it is not deemed advisable to include in this code any specific directions for making them.

RECORD OF THE TEST.

A "log" of the test should be kept on properly prepared blanks containing headings as follows:

TABLE NO. 58.

	Pressures.			Temperatures.					Fuel.		Feed-Water.		
Time.	Barometer.	Steam-gauge.	Draft-gauge.	External air.	Boiler room.	Flue.	Feed-water.	Steam.	Time.	Pounds.	Time.	Lbs. or cu. ft.	

REPORTING THE TRIAL.

The final results should be recorded upon a properly prepared blank, and should include as many of the following items as are adapted for the specific object for which the trial is made. The items marked with a * may be omitted for ordinary trials, but are desirable for comparison with similar data from other sources.

Results of the trials of a.....

Boiler at.....

To determine.....

1.	Date of trial.....	
2.	Duration of trial.....	hours.
	DIMENSIONS AND PROPORTIONS.	
	(Leave space for complete description).	
3.	Grate-surface.....wide.....long.....area.....	sq. ft.
4.	Water-heating surface.....	sq. ft.
5.	Super-heating surface.....	sq. ft.
6.	Ratio of water-heating surface to grate-surface.....	
	AVERAGE PRESSURES.	
7.	Steam-pressure in boiler, by gauge.....	lbs.
*8.	Absolute steam-pressure.....	lbs.
*9.	Atmospheric pressure, per barometer.....	in.
10.	Force of draught in inches of water.....	in.
	AVERAGE TEMPERATURES.	
*11.	Of external air.....	deg.
*12.	Of fire-room.....	deg.
*13.	Of steam.....	deg.
14.	Of escaping gases.....	deg.
15.	Of feed-water.....	deg.

	FUEL.		
16.	Total amount of coal consumed	lbs.	
17.	Moisture in coal	per cent.	
18.	Dry coal consumed	lbs.	
19.	Total refuse, dry.....pounds=	per cent.	
20.	Total combustible (dry weight of coal, item 18, less refuse, item 19)	lbs.	
*21.	Dry coal consumed per hour	lbs.	
*22.	Combustible consumed per hour	lbs	
	RESULTS OF CALORIMETRIC TESTS.		
23.	Quality of steam, dry steam being taken as unity		
24.	Percentage of moisture in steam	per cent.	
25.	Number of degrees superheated	deg.	
	WATER.		
26.	Total weight of water pumped into boiler and apparently evaporated	lbs.	
27.	Water actually evaporated, corrected for quality of steam.	lbs.	
28.	Equivalent water evaporated into dry steam from and at 212° F	lbs.	
*29.	Equivalent total heat derived from fuel in B. T. U.	B. T. U.	
30.	Equivalent water evaporated into dry steam from and at 212° F. per hour	lbs.	
	ECONOMIC EVAPORATION.		
31.	Water actually evaporated per pound of dry coal, from actual pressure and temperature	lbs.	
32.	Equivalent water evaporated per pound of dry coal, from and at 212° F	lbs.	
33.	Equivalent water evaporated per pound of combustible from and at 212° F	lbs.	
	COMMERCIAL EVAPORATION.		
34.	Equivalent water evaporated per pound of dry coal with one-sixth refuse, at 70 lbs. gauge pressure, from temperature of 100° F. = item 33 × 0.7249 pounds	lbs.	
	RATE OF COMBUSTION.		
35.	Dry coal actually burned per square foot of grate surface per hour	lbs.	
*36.	Consumption of dry coal per hour. Coal assumed with one-sixth refuse. — Per sq. ft. of grate surface	lbs.	
*37.	Consumption of dry coal per hour. Coal assumed with one-sixth refuse. — Per sq. ft. of water heating surface	lbs.	
*38.	Consumption of dry coal per hour. Coal assumed with one-sixth refuse. — Per sq. ft. of least area for draught	lbs.	
	RATE OF EVAPORATION.		
39.	Water evaporated from and at 212° F. per square foot of heating surface per hour	lbs.	
*40.	Water evaporated per hour from temperature of 100° F. into steam of 70 lbs. gauge pressure — Per sq. ft. of grate surface	lbs.	
*41.	Water evaporated per hour from temperature of 100° F. into steam of 70 lbs. gauge pressure — Per sq. ft. of heating surface	lbs.	
*42.	Water evaporated per hour from temperature of 100° F. into steam of 70 lbs. gauge pressure — Per sq. ft. of least area for draught.	lbs.	
	COMMERCIAL HORSE-POWER.		
43.	On basis of 30 lbs. of water per hour evaporated from temperature of 100° F. into steam of 70 lbs. gauge pressure (34½ lbs. from and at 212°)	H. P.	
44.	Horse-power, builders' rating, at.....square feet per horse-power	H. P.	
45.	Per cent. developed above, or below, rating	per cent.	

During the boiler trial the various observations (temperatures and pressures) should be made at the same time and at about fifteen minute intervals. In order to attain the result of the test the following must be known:

1. The amount in pounds, of coal burned and the number of pounds of ashes left..

2. The number of pounds of water pumped into the boiler.

3. The pressure of the steam in the boiler.

4. The temperature of the feed water when it enters the boiler.

5. The quality of the steam discharged from the boiler, that is, the per cent. of moisture in the steam.

There are several ways of determining the amount of water pumped into the boiler. The best method is to weigh it in tanks or barrels on scales, which have been adjusted and tested with a standard weight. The barrels or tanks should be two or more in number and of sufficient size so that the filling, weighing and emptying may not be hurried. The barrels should be placed high enough to discharge into the tank from which the feed water is drawn by the pump or injector. The valves should be large and quick opening so that the emptying will not be delayed.

If barrels are used they should be numbered and the weights of each accurately kept, and each weighed when empty and when full. When the water level in the tank has fallen so that it will hold another barrel of water, the barrel should be emptied into the tank and while emptying the other filled and weighed.

The weigher must use care and intelligence and must not become confused in his weighings or records. At the beginning of the test the level in the water tanks should be recorded, and at the end the level brought to the same mark.

If it is inconvenient to weigh the water, it may be measured either by a meter or by measurements in the tank. If a meter is used it should be tested and its error determined. The meter must be kept free from air or the readings will be in excess of the true ones.

The level in the water glass of the boiler should also be noted at the beginning and at the end of the test. A graduated scale attached to the gage glass is a convenient means of determining the water level.

The temperature of the feed water should be taken occasionally to get an average temperature.

If the boiler is fed by a steam pump, it should be run by steam taken from some other boiler if convenient; if not, the amount of steam used by the pump must be determined and allowance made.

The steam used by the pump may be condensed and the water weighed. The exhaust from the steam pump can be run into a barrel of cold water and the barrel weighed before and after.

The coal for the furnace should be weighed in barrows or barrels. It may be fired directly from the barrels or barrows or dumped on the floor before firing. The barrows should be weighed when full and empty and the time recorded in order that the weight of no portion be neglected. Any unburned coal must be weighed and the amount subtracted.

The condition of the fire should be the same at the beginning and at the end of the test. For a long test, 24 hours or more, the condition is estimated by the eye; if the test is short it is often started and stopped with bare grate bars.

Under a short test, the boiler must be in good working order about an hour before the beginning of the test so that the boiler, brickwork, etc., may be thoroughly heated. The fire is allowed to become low and by reducing the amount of steam taken from the boiler the pressure is kept up. About a half hour before beginning the test the fire is drawn and the grate cleaned. A new fire is quickly started with wood and when coal is first thrown on the test is started. Toward the end of the test the fire is allowed to burn low and at the end it is drawn out and quenched with as little water as possible. The unburned coal is picked out and its weight subtracted from the coal burned. The clinker and ashes are separated and weighed with the ashes and clinker formed during the test. The wood used is weighed and considered as equal to .4 the same weight of coal.

The temperature of the feed water can be taken best by means of a thermometer in a cup filled with oil and screwed into the feed pipe near the check valve. If the temperature is nearly constant, readings at intervals of fifteen minutes are sufficient; if it varies it should be read every five minutes.

The steam pressure shown by the gage ought to be as nearly constant as possible throughout the test and should be practically the same at both the beginning and end. Readings of the gage should be recorded every fifteen minutes and the fireman should

read it more often to keep it constant. The gage should be tested and a correction allowed for if necessary.

Barometric readings also should be taken; two or three being sufficient for a 10-hour run. Barometer readings in inches of mercury are reduced to pounds pressure by multiplying by .491; .491 pound being about the weight of 1 cubic inch of mercury.

If a vertical boiler furnishing superheated steam is being tested both a pressure gage and a thermometer should be used so that the amount of superheating may be readily found by subtracting the temperature due to pressure (obtained from steam tables) from the thermometer readings.

The quality of the steam is determined from a throttling, separator or barrel calorimeter, the first or second being preferable on account of accuracy.

Samples of flue gas may be taken at intervals from various places; from the chimney, uptake and back of the bridge. The object is to determine whether there is sufficient air admitted or whether too much air of dilution is used. The determination of the constituents which form the flue gases is made by a chemical analysis.

The draft in the chimney may be measured by a U tube partially filled with water or by a draft gage.

The above are only a few precautions to be used during the test and merely serve as a guide. Many of these readings can be taken by any careful, intelligent person, but skill in conducting a test can be attained only by experience. As the chance for error is so great, care and accuracy must be used throughout, both during the test and while working up the data.

CARE OF BOILERS.

Starting the Fire. The fireman should first ascertain the water level; as the gage glass is not always reliable on account of impurities, foam, etc., the gage cocks should be tried. In a battery of boilers, the gage cocks of each should be opened, for the water may not stand at the same level in each. The safety valve should be raised slightly from its seat. If the fire has been banked over night, open the drafts and rattle down the ashes and clinkers from the grate. In case the fire has been allowed to go out, a

new one may be started if the gage glass shows the proper amount of water and the valves work well.

If anthracite coal is used, first throw a thin layer of coal all over the grate, then place a piece of wood across the mouth of the furnace just inside the door and lay other pieces of wood at right angles to the cross piece with the ends resting on it. This allows a space under the wood for air. Now throw on coal until the wood is covered. The fire may be started with oily cotton waste, shavings, or any combustible material.

Keep the furnace door open and the draft plate closed until the wood is burning freely, which causes the flame to pass over and through the coal and to ignite it. The fire is then spread or pushed back evenly over the furnace bars; the furnace door closed, the ash pit door opened, as the draft requires, and more coal added when necessary. If bituminous coal is used, do not spread a thin layer over the grate bars under the wood.

The fire at the start should be slow, to cause gradual, uniform heating of the water and various parts of the boiler. If steam is raised too rapidly, enormous strains are set up, due to unequal expansion, thereby causing leakage at joints and perhaps rupture.

If the boiler is of the water tube type, steam may be raised more rapidly, because the amount of water is less and the joints are usually placed at some distance from the intense heat of the fire.

The fire being started, the method of adding coal depends upon the fireman, the kind of coal, the type of boiler, and the rate of combustion. There are three general methods of firing, — spreading, alternate or side firing, and coking.

Spreading is accomplished by placing small amounts of coal uniformly over the entire surface of the grate at short intervals. By this method, the coal is thrown just where it is wanted and then not disturbed. The fire should be hollowed in the center, that is, it should be thicker at the sides. Good results are obtained from this method, since the fire can be kept in the right condition at all times, if the coal is of the right sort. During the operation of firing, the door should be kept open as little as possible or the fire will be cooled by the entrance of the cold air. For a short time, while the coal is giving off gas, the draft plate of the furnace

door should be opened, in order that sufficient air may be admitted above the coal to burn the hydrocarbons.

When the **alternate** or **side firing** method is used, coal is spread so as to completely cover one side of the fire at one firing, leaving the other side bright. At the next firing, the bright side is covered. The hydrocarbons given off by the fresh coal are burned by the hot gases from the incandescent coal. This method is superior to spreading, because the entire furnace is not cooled off by the addition of fresh fuel.

Side firing is most advantageous with two furnaces leading to a common combustion chamber. The furnaces are fired at regular intervals with moderate charges of coal, and the draft plates are opened while the coal is giving off gas.

The two systems described above are best adapted to anthracite coal, since it burns with comparatively little smoke.

With bituminous coal, which is soft and burns with considerable smoke, the **coking system** is used. The coal is piled on the grate just inside the door, and allowed to coke from 15 to 30 minutes. During this time, the hydrocarbons are driven off and burned by the heat from the fire. In order to fully accomplish this, air must be admitted above the grate through the draft plates of the furnace door. The coke is then pushed backward over the fire and a new supply placed on the front of the grate. The air admitted prevents the forming of carbon monoxide gas and smoke. At the same time, it cools the furnace somewhat and reduces the rate of evaporation. This objection is not serious, unless a boiler must be worked to its fullest capacity in order to furnish the required amount of steam. If this is the case, economy is sacrificed to rapidity, for a low rate of combustion is usually more economical than a high rate. The necessary thickness of a bed for the best results is found by experiment. It depends on the draft and the kind of coal used. If the former is strong and the coal in large lumps, it may be thick, (about one foot) ; but if the draft is weak or if the coal is small, the bed must be thin (about three or four inches) so that sufficient air may pass through. In marine and locomotive work, with forced draft, the bed must be very thick to get a large coal consumption per square foot per hour.

With the same draft, bituminous coal may be fired more thickly than anthracite.

After finding from experiment the best thickness for the bed, keep it at that thickness. Always keep the bed of uniform thickness and never let the fire burn holes in the bed, and do not let the rear of the grate become bare. If a larger amount of steam is required, fire smaller quantities at more frequent intervals. Do not fire a large amount of coal and wait for the pressure to rise. The firing of fresh coal chills the furnace and temporarily retards combustion. It should be fired in small quantities and as quickly as possible. Keep the fire free from ashes and clinkers, but do not clean the fires oftener than is necessary.

Four tools are used for cleaning the fire. The slice-bar, the prick-bar, the clinker hook, sometimes called the devil's claw, and the hoe or rake.

The slice-bar is a long straight bar with the end flattened. It is used to break up clinkers by thrusting it between the grate and the fire. It is also used to break up caking coal. The prick-bar is similar to the slice-bar except that the end is bent at right angles like a hook. To remove ashes, the prick-bar is run along up between the grate bars from underneath. This bar is often made with detachable hook, so that the end may be replaced when burned off. The clinker hook, or devil's claw is used to haul the fire forward. The hoe, or rake, is used to draw out cinders, to haul the fire forward, etc.

In cleaning the fire, the fireman first looks to the water and steam. There should be enough water and sufficient steam pressure to last during cleaning. Then he breaks up the clinkers with the slice-bar and removes the ashes with the prick-bar. If it is necessary, he pushes the fire to the rear, thoroughly cleans the front of the grate bars, then hauls it forward and cleans the back of the furnace bars. Some firemen clean one side at a time instead of first the front and then the rear. The fire should be allowed to burn down before cleaning, but sufficient fuel, called chaff, should be left to start the fire quickly. Before cleaning, partly close the dampers so that the amount of cold air admitted will be small. For this reason and to prevent loss of pressure, clean as rapidly as possible.

Banking the fire depends upon the condition of the fire, the fireman himself and the length of time it is to remain banked. First clean and place all the coal in a small space at the bridge, then cover with fresh coal to a depth depending on the length of time the fire is to remain banked. Then close all dampers and open the door. Some firemen cover the front of the furnace bars with ashes.

To start from a banked fire, first examine the condition of the water level, steam pressure, safety valves, etc. Then clean the fire with the slice-bar and rattle down the ashes with the prick-bar. After spreading the coal evenly over the grate cover with a thin layer of coal and open the dampers.

Condition of Water. The first duty of the engineer or fireman is to know at all times the height of water in each and every boiler under his care. The gage cocks should be tried frequently, because the water gage is not always reliable. The fires should never be unbanked nor replenished without sufficient water in the boiler.

Gage Cocks and Water Glasses should be kept clean and in constant use. This is important because if the connections become clogged by dirt and impurities, the gage glass will not show the true level.

The Safety Valve. These valves should be of ample size and in good working order. They should be raised cautiously and frequently, as they are likely to stick to their seats if seldom used, or if the feed water is corrosive. They should be tried at least once a day.

The Pressure Gage. The pointer of the pressure gage should stand at zero when there is no steam pressure in the boiler. If it does not, it should be corrected and compared with a standard gage. If the gage indicates the limit of pressure and the safety valve does not blow off, bank the fire; also if convenient, test the gage and if found to be correct, investigate the condition of the safety valve.

Fusible Plugs. Whenever the boiler is cleaned these safety devices should be cleaned on both sides. They should be scraped if necessary. If not kept clean they are likely not to act.

The Ash Pit. Keep the ash pit clean to prevent the excess-

Standard Belt-Driven Pump.

Direct-Connected Steam Pump. (Special.)

Standard Belt-Driven Pump.

ive heat underneath from burning the grate bars. If the ash pit is shallow, it is often filled with water to a depth of about two inches.

Cleaning. All parts of the boiler which are exposed to the fire should be kept clean. All tubes and flues should be swept as often as possible. The frequency of cleaning depends upon the kind of fuel and the nature of the water. No boiler is economical if the heating surfaces are allowed to become covered with a thick bed of soot and dirt. The layer of dirt should never be over $\frac{1}{16}$ of an inch in thickness. Handholes and manholes should be frequently removed and surfaces examined and cleaned. In replacing handhole caps and manhole covers, clean the machined surfaces without scratching, smear with oil and screw up tight. The boiler should be emptied as often as once in two weeks and even more often if the feed water is impure. The surface blow-out should be opened when there is no demand for steam. If the water is muddy, or comes from a condenser and contains oil, five or six inches of water should be blown out every day. Never blow off when the steam pressure is high. Make sure no water is escaping when the blow-off cock is supposed to be closed. Scale should be prevented if possible, but if it is formed, it should be removed frequently as it is a poor conductor of heat, thereby reducing the efficiency of the heating surfaces and rendering them likely to burn. Scale may be removed by any one of several methods, but those which require sharp tools, or which cause unequal expansion should never be used, as the boiler may be seriously damaged.

Feed Apparatus. It is good practice to have two independent ways of feeding the boiler. The pump and the injector should both be of sufficient capacity to supply the feed water and whichever is in use should be run as uniformly and continuously as possible. Check valves and feed valves should be examined and cleaned frequently. Cold water should never be fed into any boiler when it can be avoided. But if it is necessary to do so, it should enter the boiler so that it will not come in contact with the plates before it has mixed with the warm water in the boiler.

Low Water. In case of low water, cover the fire with wet ashes, or fresh coal if ashes are not convenient. Draw the fire as soon as may be without increasing the heat. Do not admit

more cool water. Do not start or stop the engine or lift the safety valve. Never fill a hot boiler with cold water as leaks and fractures may result.

Foaming. Close the throttle or stop valves and keep it closed until the water becomes quiet and shows its true level. Foaming, if caused by dirty water, can usually be stopped by blowing off and feeding fresh water. If the boiler primes, lower the water line.

Blowing Off. Never empty a boiler while it is hot, because the deposits of sediment and grease will become baked on the heating surfaces and will have to be removed by mechanical means. One of the best methods of preventing scale is to allow the boiler to cool gradually and stand a few days with cold water in it before emptying.

Dampness. The exterior of boilers should be examined frequently, especially if the location is damp, or there is much brickwork setting. No water should come in contact with the exterior of the boiler, either from leaky joints or other sources, as it causes corrosion. Do not allow the covering to become damp.

Air Leaks. The admission of air from the brickwork to the combustion chamber, or to the flues is a source of loss.

The boiler should often be examined for blisters, cracks, and defective joints, and if any appear, should receive prompt attention.

Galvanic Action. If galvanic action between the water and the copper and iron is suspected, hang a piece of zinc in the boiler.

If a boiler is to stand unused for some time, it may be kept in good condition by any one of several methods. It may be completely filled with water in which a quantity of common soda has been dissolved, or it may be emptied, thoroughly dried, and trays of quicklime placed in the bottom. The boiler is then sealed as nearly air tight as possible.

SECTION 6.

THE FLYING-MACHINE.

Ever since or even before the day when Dædalus and his son Icarus made a flight through the air on wings of wax and came to grief because of a too near approach to the sun, it has been the ambition of men to fly. Centuries came and went, and still men remained on the surface of the sea or land until, in 1772, the Montgolfier brothers invented the first balloon and made an ascension from Annonay, in France, on June 5, 1783.

Their balloon was made of silk and was inflated with hot air. It was a bold adventure, but its success was its own warrant, and others followed who improved upon the original construction until now ballooning has come to be looked upon as an every-day performance and an adjunct to the attractions of county fairs and a means of exploitation for the circus.

In general terms it may be asserted that any gas that is lighter than the air can be used for the inflation of the balloon. Where high altitudes are to be reached, hydrogen, as the lightest of the gases, is used; but where the ascension is for the mere purpose of amusing a multitude, common illuminating gas or even hot air is employed.

The advantage of using hydrogen gas lies in its extreme lightness, since it weighs only about .07 as much as atmospheric air, whereas coal gas weighs .4; while heated air is subject to constant radiation, and must receive a continual supply of heat from an external source in order to maintain its state of rarification, else the ascension will come to an abrupt termination. Assuming that the specific gravity of the air is .001205, these figures show that for each pound of load, including the weight of the balloon, attachments, car and aëronaut, a balance will be obtained by the use of about 14.25 cu. ft. of hydrogen gas, or, if a buoyancy of two to one is to be obtained, this quantity must be doubled, using 28.5 cu. ft. per pound of load.

With coal gas, weighing four-tenths as much as the atmosphere, the balance is obtained by the use of about 22.13 cu. ft.,

or a twofold buoyancy with 44.26 cu. ft. per pound of load, an increase of nearly 56 per cent over the requirements when hydrogen is used. This very materially increases the volume of the balloon and renders the use of illuminating gas out of the question for very high ascensions.

The best balloons are made of silk, though cotton is used for short ascensions. In making passenger balloons the material is first cut into strips and sewn together. The globe thus formed is then partially filled with air and given several coats of varnish, when it is tested for leaks by covering with water; the air inside appearing as bubbles at every leak.

At the top a valve is fitted to permit the escape of gas when it is desired to descend, and over all a strong network is placed, to which the car is attached.

The smallest balloons intended for carrying passengers usually have a capacity of at least 12,000 cu. ft., and weigh about 280 lbs.

With balloons of this character some exceedingly high ascensions have been made. One of the most notable was that of Professor Berson from Berlin, Germany, on Dec. 4, 1894.

"Professor Berson's last reading showed the barometer 9.12 in. and a temperature of —54° F. If we consider that all of the errors in the barometer and thermometer had been allowed for, and also that the temperature of the earth was 37° and air pressure 30.02 in., we have a height of about 28,750 ft. as the highest made in this voyage." Such heights as this are only possible when a supply of oxygen is carried to maintain life.

The balloon, however, is not, strictly speaking, a flying-machine, nor is it ordinarily under any control as far as its horizontal direction is concerned. It moves with the wind and, to the aëronaut, is in a calm at all times.

The first step in advance of the drifting balloon is to be found in one fitted with some form of propelling machinery, by which the movement is controlled, the balloon navigated and made to move to a certain extent independently of the direction and velocity of the wind.

The difficulties lying in the way of a satisfactory solution of this problem are very great. These are: The great bulk of the supporting balloon, whereby a corresponding area is presented to the action of the wind; and the necessity for extreme lightness and great power in the motor.

As the first ballooning was done in France, so that country has maintained the lead in aëronautical matters to the present time. There is a regular balloon corps attached to the army, and for many years especial attention has been directed to the design and construction of dirigible balloons. The details of the mechanisms

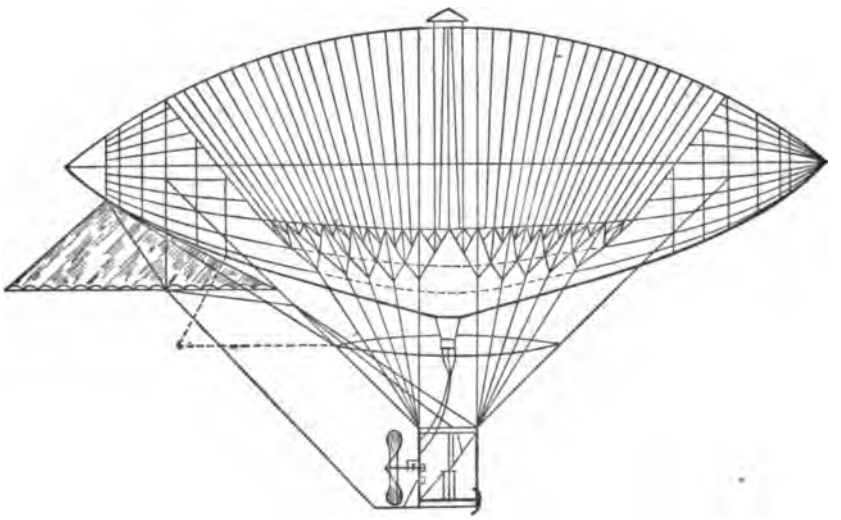


Fig. 1. M. Gaston Tissandier's Dirigible Balloon.

employed, as well as of the exact contour of the gas envelope, are guarded as military secrets, but enough is known to give a general idea of what has been done.

Attention was first publicly directed to the success attained with the dirigible balloons of the French army in 1893, by an article in the *Mémoires de la Société des Ingénieurs Civils*.

The true progenitor of the dirigible balloon was General Meusnier, who, in a series of papers published in 1783, set forth a scheme for a dirigible balloon. It was not, however, until 1852, that Henri Giffard made a rational attempt to construct such a one. His balloon was spindle-shaped, and measured 144 ft. 4 in.

from point to point. His motor was, however, too weak to permit of making any headway against even a moderate breeze, though the balloon could be steered and made to describe circles while drifting. His general form of envelope construction has been followed by all experimenters since that time.

In 1870 M. Dupuy de Lome undertook the task of constructing a dirigible balloon. The propeller used was two-armed, 19 ft. 8 in. in diameter, and was turned by eight men. The spindle-shape was used for the envelope.

A trial was made at Vincennes in 1872, where in a breeze of 39 ft. per second a deviation of 12° was obtained. The stability was perfect despite the exertions of the eight men at the crab; still the balloon was a drifting one, and was dirigible only in the sense that it could be made to deviate from the true course of the wind. To an outside observer the problem seemed as far from solution as ever, but when taking into consideration the insufficiency of his motor, the committee appointed to be present at the trial said: "It serves as a starting-point for all who wish to continue in this direction."

Others have desired to continue in this direction, and the first to achieve results that even approached success was M. Gaston Tissandier, who, with his brother Albert, in 1884, constructed the balloon illustrated in Fig. 1.

It was fitted with a Siemens motor driven by a bichromate of soda battery that was very ingeniously arranged so as to minimize the weight and, at the same time, produce the greatest possible effect. The motor weighed 121 lbs. and the cells 496 lbs., and they contained liquid enough to work for two hours and a half, generating during that interval $1\frac{1}{3}$ horse-power. The screw had two arms and was a little over 9 ft. in diameter.

Several ascensions were made with this balloon. In one, undertaken in 1884, the motor developed $1\frac{1}{2}$ horse-power and was sufficient to propel the structure through the air at a rate of thirteen feet per second. After having practically followed the direction of the wind, which had a velocity of about ten feet per second, during which the rudder turned it aside a little, the balloon was made to describe a semicircle, and brought up with its head into the wind, where it was navigated for about ten minutes

directly above Grenelle, and the same evolution was repeated above the observatory. Through the exhaustion of the battery it became impossible to return to Paris.

These experiments were quickly followed by the work of Commandant Renard of the French balloon corps. With a spindle-shaped balloon, but with a more powerful motor, and the screw placed at the front instead of the rear, he succeeded in making several ascents and returning to the point of departure. The motor developed $8\frac{1}{2}$ horse-power, with a weight of 220.5 lbs.

In an ascension made on Sept. 23, 1885, the balloon started from Chalais-Meudon against the wind and went to Paris, where

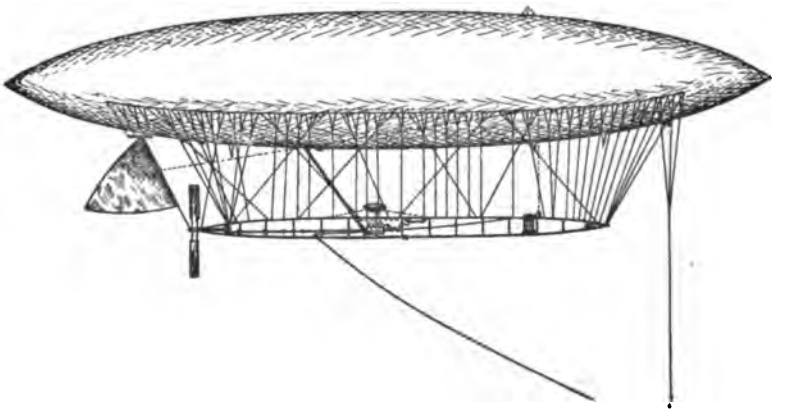


Fig. 2. Dirigible Balloon of Santos Dumont.

it was easily handled, and then returned to Chalais. The average speed throughout the trip was 21.33 ft. per second.

This work has been continued by the French balloon corps, until now the army is possessed of a dirigible balloon that may be considered controllable in light airs.

The problem has not, however, been left solely to the military departments of the several nations of the earth for solution, for private investigators have been constantly at work upon it. In 1901 a young Brazilian named Santos Dumont made a series of brilliant experiments in France, constructing several balloons, one after the other, each being an improvement on its predecessor. Fig. 2 shows the sixth of the series, which was known as the Santos Dumont VI.

In striving for a prize of $20,000, offered by M. Deutsch for the first dirigible balloon that should start from the Parc d'Aérostation at St. Cloud and be sailed around the Eiffel Tower in Paris and brought back to the starting-point in thirty minutes, he succeeded in covering the ground in but forty seconds above the stipulated time. The total distance is a little less than nine miles.

This balloon had a length of 108 ft., a diameter of 19 ft. 6 in., and was driven by a 4-cylinder petroleum motor weighing 216 lbs., and developing 20 horse-power. The screw was a little more than 13 ft. in diameter, and was turned at a speed of 300 revolutions per minute.

Instead of the old-fashioned network, the basket and machinery were suspended direct from the envelope by strong piano wire. Japanese silk was used for the envelope itself, and was made impervious to the gas by four coats of linseed oil.

The problem of the navigation of the air is thus seemingly resolved into the development of a motor which shall be so light that it can be carried in the car of a balloon inflated with hydrogen gas, and which shall develop sufficient power to propel the whole apparatus at a speed that should not be less than forty feet per second, or about fifteen feet faster than the time made by M. Santos Dumont in 1901, whose speed during his most successful trip was only $1\frac{1}{2}$ ft. faster than that made by Commandant Renard in 1885.

That these conditions of greater power and less weight can be met, there can be no doubt, as will be shown in the discussion of the true flying-machine.

We have seen that the balloon is a mere drifter in the air, and that even when fitted with propelling and steering machinery there is still much to be done before it can be said to be truly controllable under all ordinary conditions.

But even when so built as to be "navigable" in the broad application of the word, it is not a "flying-machine" in the true sense of the term.

Prof. S. P. Langley of the Smithsonian Institution, who has probably done more than any other one man in the investigation of aëro-dynamics, defined a "flying-machine" as "something much

heavier than the air, and entirely different in principle from the balloon," and which is dependent, like the bird, upon its own power and the resistance of the air for its motion and support. The true flying-machine, then, is "in no way concerned with ordinary aëronautics, or the use of balloons, or objects lighter than the air, but solely with the mechanical sustentation of bodies denser than the air."

It will be seen that the flying-machine differs as widely as it is possible from the dirigible balloon. Innumerable attempts have been made to build flying-machines, but, with one exception, they have been failures.

The soaring bird has been a source of curiosity for years, and the mystery of its rising into the air on motionless wings has not

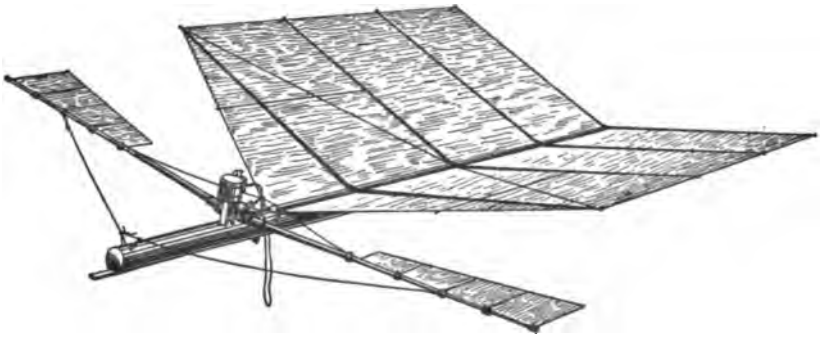


Fig. 3. Hargrave's Flying-machine.

as yet been unraveled. The one thing certain is that some motion is needed before the soaring can begin. To quote from Professor Langley: "Some soaring birds do this by an initial run upon the ground, and even under the most urgent pressure cannot fly without it. Take the following description of the commencement of an eagle's flight on the sandy banks of the Nile: An approach to within eighty yards aroused the king of birds from his apathy. He partly opened his enormous wings, but stirs not yet from his station. On gaining a few feet more he begins to *walk* away, with half-expanded but motionless wings. Now for the chance,—fire! A charge of number three from eleven bore rattles audibly upon his densely feathered body; his walk increases to a run, he gathers speed with his slowly waving wings, and eventually leaves the

ground. Rising at a gradual inclination, he mounts aloft and sails majestically away to his place of refuge in the Libyan range, distant at least five miles from where he rose. The marks of his claws were traceable in the sandy soil, as at first with firm and decided digs he forced his way, but as he lightened his body and increased his speed with the aid of his wings, the imprints of his talons gradually merged into long scratches. The measured distance from the point where these vanished to the place where he had stood proved that, with all the stimulus that the shot must have given to his exertions, he had been compelled to run full twenty yards before he could raise himself from the earth."

Other soaring birds rise by heading into the wind and thus gain at once an initial velocity by the motion of the wind itself. In short, the support afforded to the soaring bird and the kite is the same in principle,— the impact of moving air on the lower side of an inclined plane. And it is upon this principle that the experimenters in flying-machines have based their constructions.

By "experimenters in flying-machines" it is not intended to include those wild theorists whose efforts are wilder than their thoughts, and who plunge along in the dark, ignorant of the very element which they hope to conquer, but scientific men like Hargrave in Australia, Maxim in England, Lilienthal in Germany and Langley in America, who appreciate the difficulties before them and who make a careful study of the principles of flight before undertaking the construction of an operative machine.

It is quite impossible within the limits of the space available to do more than indicate what these leaders in the work have done.

Mr. Lawrence Hargrave of New South Wales entered upon the investigation of aërial flight a number of years ago with an extended series of experiments in kites, developing several interesting forms and obtaining a wide range of experience in the support of heavy bodies by means of this form of aëroplane.

Acting in accordance with the data that he had gathered, he built the machine shown in Fig. 3. It consisted of a central backbone which carried "two outstretched wings, or aëroplanes, which were to glide over the air, while in front were two flapping wings, intended to afford the propelling power. These wings were driven by an engine, whose motive fluid was compressed air stored

in the hollow backbone of the machine." An actual flight was obtained with this machine and on one occasion the distance covered was 343 ft. in twenty-three seconds.

Encouraged by the success of this machine, Mr. Hargrave built one with a Serpollet boiler and a steam engine. The total weight of this apparatus was 64½ ounces and it developed .169 horse-power. The machine, however, could never be maintained in long flight on account of the difficulty of balance.

The most ambitious effort in the way of constructing a flying-machine was made between the years 1892 and 1894 by Mr. Hiram S. Maxim in England. Strictly speaking, the machine was never intended to fly, but to demonstrate the possibility of such flight. It is shown in Fig. 4, and was very large and heavy. The aëroplane, which sheltered it like a roof, was 110 ft. long and 40 ft. wide, and was made of steel frames covered with silk. Other similar planes attached to this ran the total surface up to about 5,500 square feet of area. The whole apparatus, including the steering gear, was 145 ft. long. The power was developed by a 300 horse-power steam engine driving two screws at the back, and the total weight of the machine in working order with fuel and water was between 5,000 and 6,000 lbs. It was mounted upon wheels and traveled on a track of twelve feet gauge about half a mile long. On each side and two feet above the carrying rails there was an inverted track of strong timber. This was to hold the machine down to earth in case it manifested a disposition to rise under the influence of its immense aëroplane when in motion.

The only results obtained with this machine were to prove that the aëroplane was sufficient to lift it from the carrying rails and cause its wheels to run along the bottom of the inverted track, a fact that had previously been demonstrated mathematically. In fact, this flying-machine was too large to be trusted to make a true flight, as balance and the maintenance of equilibrium were out of the question, with the construction that was used.

While not strictly speaking a flying-machine in the sense of being fitted with propelling machinery, the work done by Herr Otto Lilienthal near Berlin in Germany is deserving of attention.

Herr Lilienthal was a mechanical engineer and constructor, a prominent member of the German society for the advancement of aërial navigation, and withal a good deal of an athlete. He did not enter the field with a design for a machine with a motor, but fitted himself with a pair of wings, as shown in Fig. 5, to which a rudder in the form of a tail was attached. The area of the sustaining surface was 107 sq. ft. at first, but this was afterwards reduced to 86 sq. ft. "The framework was made of willow and covered with sheeting. The weight of the whole apparatus was 39.6 lbs.

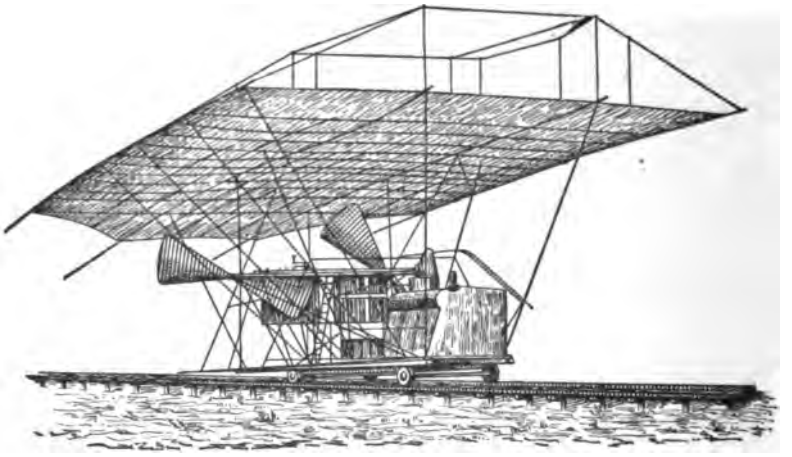


Fig. 4. Maxim's Flying-machine.

"In order to become accustomed to the management of these artificial wings, Herr Lilienthal first practised in his garden. Here he had a springboard, toward which he ran for a distance of about 26 ft.; and with the velocity thus acquired, together with the reaction of the springboard, he launched himself into the air, where he could learn to operate and manage the wings.

"After these preliminary experiments had given him dexterity and facility in the management of the apparatus, he betook himself to a hilly region in the suburbs of Berlin, and there practised soaring flight in natural winds of moderate velocities. The plan consisted in first running against the wind, and thus deriving therefrom the necessary sustaining pressure.

"Having selected a hill whose downward inclination faced

the prevailing wind, he ran along the summit straight toward the wind, until a sufficient velocity was attained at the brow, where he was carried into the air and landed safely at the foot of the hill, having sailed a distance of from 65 to 80 ft.

"When the wind velocity was greater than eleven to thirteen miles per hour, the management of the apparatus became exceedingly difficult, and Herr Lilienthal advises an experimenter not to venture to leave the ground under such circumstances, unless he has attained, through long practice, a considerable degree of dexterity in manœuvre."

Working along these lines, Herr Lilienthal succeeded in so improving the shape of his wings that the length of his flight was greatly increased. Writing on the subject of his work, Herr Lilienthal said:

"When we look at the safe and quiet sailing of the birds, it almost seems as if some undiscovered mechanical principle were at work, some feature in the elastic properties of the air or in the elastic curvature of the feathers, which accounts for the mystery of sailing flight; but my experiments have taught me that there is no mystery. Dexterity alone, in my opinion, invests the native inhabitants of the air with superiority over man in that element."

With practice, Herr Lilienthal's own dexterity was increased until he was able to make flights of a thousand feet, and that, too, not only along a line of continuous descent, but he was actually able to soar upward, and thus increase the distance between the point of departure and alighting.

Unfortunately he was emboldened by his own success and skill, and in one of his flights a sudden gust of wind disabled his apparatus and he was thrown to the ground and killed. But he had accomplished enough to warrant his own assertion that the birds owe their superiority in the air to skill. Of this skill any one can readily convince himself by watching a soaring bird at close range, when he will observe the same sudden twitching of the wings when the bird adjusts its balance in the air, as is to be seen in the movements of the gymnast when he maintains his balance on a tight-rope through similarly quick motion of his arms.

Returning now to the flying-machine, one final and successful

example will be given, that of Prof. S. P. Langley of the Smithsonian Institution.

Professor Langley has approached the problem with greater care and more elaborate preparation than any other man. For years before he undertook the construction of even the crudest of flying-machines he conducted an elaborate series of experiments upon a whirling table in which the supporting action of the air upon almost every conceivable shape of aëroplane, and at all possible velocities, was accurately measured and recorded. As his data, thus obtained, accumulated, he supplemented them by experiments with small models acting freely in the air; sometimes they

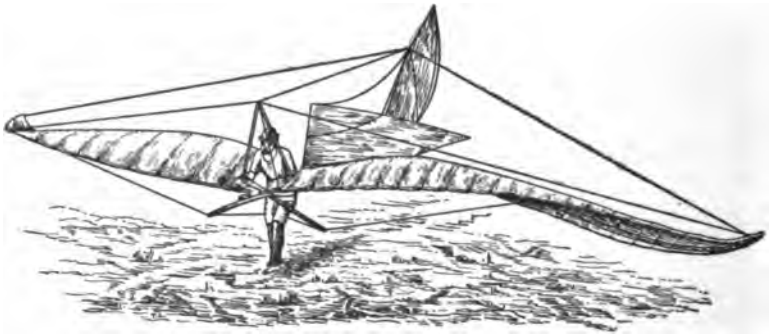


Fig. 5. Wings of Herr Otto Lilienthal as Adapted to Soaring.

were mere planes gliding through the air, and sometimes machines driven with screws propelled by rubber bands under tension.

Having satisfied himself of the possibilities of mechanical flight, he calculated the areas of sustaining surfaces that he would need, and the best shape to give them. Then came a long and elaborate series of investigations as to the motor best suited for the work, accompanied by the construction of a number, which were tested, weighed and found wanting. His final choice was the steam engine, supplied with steam from a plain copper coil for a boiler, through which a circulation was artificially maintained, and which was heated by a gasoline flame from a special jet.

After years of careful and painstaking investigation of this sort, and after building several unsuccessful machines, he constructed the one shown in Fig. 6, by which mechanical flight was made an accomplished fact.

In the completed form there were "two pairs of wings, each slightly curved, each attached to a long steel rod, which supports them both and from which depends the body of the machine, in which are the boilers, the engines, the machinery and the propeller wheels, these latter being not in the position of those of an ocean steamer, but more nearly amidships. They are made of wood, and are between 3 and 4 feet in diameter.

"The hull itself is formed of steel tubing; the front portion is closed by a sheathing of metal, which hides from view the fire-grate and apparatus for heating, but allows us to see a little of the coils of the boiler and all of the relatively large smokestack

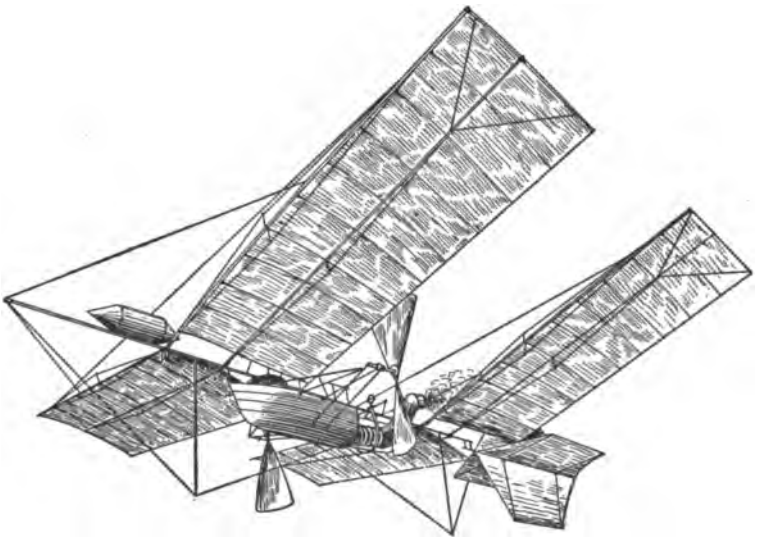


Fig. 6. Langley's Flying-machine.

in which it ends. The conical vessel in front is an empty float, whose use is to keep the whole from sinking if it should fall in the water.

"The boiler supplies steam for an engine of between 1 and 1½ horse-power, and with its fire-grate weighs a little over 5 lbs. This weight is exclusive of that of the engine, which weighs, with all its moving parts, but 26 ounces. Its duty is to drive the propeller wheels at a speed of from 800 to 1,200 revolutions per minute."

The rudder has both a horizontal and vertical blade, so as to steer in both directions, and is arranged to perform this function automatically.

The total length of the machine is about 16 feet, and the span of the wings from tip to tip is between 12 and 13 ft., while the weight of the whole, including the machinery, is nearly 30 lbs.

Of course this machine cannot be regarded as more than a model, but, such as it is, it is an operative one and complete in itself.

On the day when it made its first successful flight from a houseboat on the Potomac, at its first launching it made a short, sharp dive into the waters of the river. No change was made other than a slight adjustment of the conical float at the front, whereby the center of gravity of the whole machine was moved about three-quarters of an inch. Apparently this was sufficient to obtain the desired balance. That is the crucial point to be met in all flying-machines. The second launching resulted in a flight of a little more than half a mile at a speed of between 20 and 25 miles an hour, when the machine again fell into the river through the exhaustion of its supply of fuel and water. A renewal of these supplies was made, and again the machine soared through the air like a thing of life. Taught by this experience, another machine of similar construction was adjusted and made a flight, on a later day, of about three-quarters of a mile at about the same speed, thus fixing true mechanical flight as an accomplished fact.

There can be no doubt that it lies quite within the powers of man to construct a machine of such power and wing area that heavy bodies may be transported through the air at high velocities. This is not the trouble.

All experimenters are agreed that it is the maintenance of equilibrium that is the difficulty. Lilienthal said that the bird owes its power of flight to its dexterity. Hargrave's machine was a failure because it was not properly balanced. Maxim dared not trust his great machine off the surface of the earth, and Langley achieved success after years of patient labor only by securing the proper balance.

SECTION 7.

THE GAS ENGINE.

About fifty years ago many ingenious air engines were invented but all except those of small size were unsuccessful. The principal reason for their non-success was the difficulty with which heat is given to and taken from air. Water, on the other hand, readily absorbs heat and becomes steam which just as readily gives up its heat and again takes the form of water. A gas engine may be considered as being an air engine in which the difficulty stated above has been partly overcome by mixing with the air a small portion of gas, usually either ordinary coal gas used for illuminating, or some cheaper gas made especially for the purpose.

When air alone is the working substance, it gets its heat from an external furnace in the same manner as the working substance (steam) of a steam engine receives heat from the furnace of a boiler. These are called *external combustion engines.*

In another class of engines, the combustion which supplies heat takes place in a closed cylinder which contains the working substance. The combustible gas or vapor is mixed with air in the cylinder of the engine and then ignited. The gas combines so rapidly with the oxygen of the air that an explosion takes place. The pressure thus suddenly created drives the piston in the cylinder. Such engines are called *internal combustion engines.*

In the steam engine, using a saturated vapor as the working substance, the pressure and temperature are dependent upon each other, that is, if the temperature is high, the pressure also is high. In other words, a high temperature cannot be obtained without great pressure, unless excessive superheating is resorted to. In the gas engine it is possible to use an initial temperature much higher than the limit in a steam engine, because the temperature and pressure of the working substance are independent of each other. It is a well-known fact that the efficiency is proportional to the difference between the initial and the exhaust temperatures, therefore, if the lower limit is not raised correspondingly, the effic-

ency of a gas engine will be much greater than that of an ordinary steam engine.

In the external combustion engine, there is a considerable drop in temperature; the temperature in the furnace being much greater than that of the working substance (air or steam) in the engine cylinder. The internal combustion engine possesses this advantage: The temperature which is produced by combustion in the cylinder is the upper limit of the thermodynamic cycle.

Types of Gas Engines. One of the first practical gas engines was that introduced by Lenoir in 1860. In general appearance it resembled a small horizontal steam engine. The diagram for one revolution is shown in Fig. 1. During the first portion of the stroke, the admission valve (a slide valve) was open

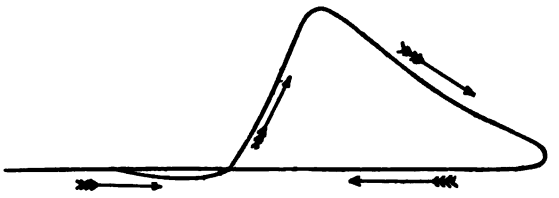


Fig 1.

and air and gas were drawn into the cylinder at atmospheric pressure. As the admission valve began to close the pressure fell slightly, because the air and gas could not enter the partly closed valve fast enough to fill the cylinder at atmospheric pressure; this caused the line to fall below the atmospheric line. At about half-stroke the admission valve closed and the charge was ignited by an electric spark. The explosion caused a sudden rise in temperature and pressure; the heated products of combustion then did work on the piston during the rest of the stroke. During the back-stroke, the products of combustion were expelled from the cylinder. While the piston was returning and driving out the products of combustion, admission, ignition, explosion, etc., were taking place on the other side of the piston. In other words the engine was double acting. In order to prevent the cylinder from becoming excessively heated, water was kept circulating in a jacket which encased the cylinder.

This engine was not economical; it used about 95 cubic feet

of gas per horse-power per hour, or more than four times as much as an ordinary modern gas engine. This was partly due to the fact that it did not use compression.

In 1866, Otto and Langen invented an engine which was in some respects a great improvement, although it did not compress the charge. The engine was vertical and the explosion took place under the piston, which caused it to rise rapidly. When the piston reached the end of the stroke the gases had become cooled and the pressure was below atmospheric. This caused the piston to descend, and as it fell it became geared to the shaft.

This engine used only about 40 cubic feet of gas per horse-power, but it was noisy and irregular in action. A few years later it was displaced by an engine that was more like the Lenoir, but in the new engine the gases were compressed before ignition.

Beau de Rochas and Otto. The French scientist, Beau de Rochas, first proposed the chief features of the modern gas engine. He first pointed out the advantage of compressing the explosive charge before ignition, in order to get a higher temperature and a greater expansion. He proposed to have four operations take place during two revolutions or four strokes.

BEAU DE ROCHAS CYCLE.

First Revolution.

First, the forward or suction stroke; during this whole stroke the charge of gas and air is drawn into the cylinder.

Second, the backward or compression stroke; during this stroke the charge is compressed in a large clearance space in front of the piston.

Second Revolution.

Third, the forward or power stroke; ignition takes place at the beginning of this stroke (or at the dead point) and is followed by explosion and expansion.

Fourth, the backward or exhaust stroke; during this stroke the burned gases are discharged from the cylinder.

This "four-stroke cycle" (see Fig. 2) is now used in most all gas engines. Beau de Rochas also suggested high speed and small cylinder surface but these valuable suggestions were not applied successfully until Otto re-invented the four-stroke cycle.

Although the cycle is usually called the "Otto," the invention belongs to Beau de Rochas.

The Otto Engine. Most of the gas engines of to-day closely resemble the "Otto," hence a brief description of this type will give the reader a general idea of the action and construction of the gas engine. In the "Otto" engine, the cylinder is single acting, that is, the explosion takes place only on one side of the piston. Before ignition the explosive mixture is compressed in the clearance space, which is usually about one-half the volume swept through by the piston. Cold water circulating in a jacket

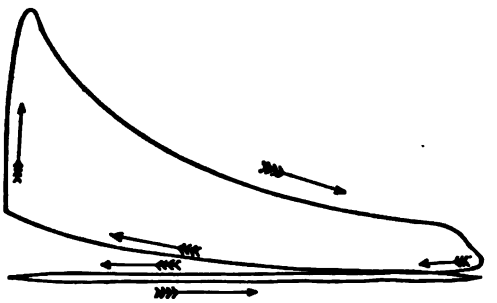


Fig. 2.

keeps the cylinder cool so that it may be lubricated. The length of the piston is about $1\frac{1}{2}$ times the diameter; on account of its length it slides without binding. It is made tight by split packing rings which press against the cylinder walls. The connecting rod is fastened directly to the piston by the wrist pin, and to the crank by the crank pin. No crosshead is required. A fly wheel and pulley are keyed to the crank shaft; the fly wheel to make the speed more regular, and the pulley for the transmission of the power.

On the shaft is placed a bevel gear which meshes with another bevel gear on the countershaft or valve shaft. The countershaft is parallel to the center line of the engine and makes one revolution to two of the crank shaft; this is accomplished by giving the bevel gear on the countershaft twice as many teeth as the bevel gear on the crank shaft. This ratio is necessary because the valves are in a given position but once during the cycle (two revolutions). In some engines the bevel gear is replaced by a

screw gear. In the older engines a slide valve controlled the admission of the charge to the cylinder and also the time of ignition. The slide valve is operated by a rod connected to a crank on the countershaft. The exhaust valve (conical seat) is opened by a cam moving a lever whose fulcrum is near the center; when the cam pushes the end of the lever down, the other end rises and opens the valve which is kept closed by a spring.

The speed is controlled by a centrifugal governor; when the balls rise (due to increase of speed) a roller that bears against a cam is thrown out of contact with the cam, which prevents the gas valve from opening. Thus, no gas is admitted. In this way the engine misses an explosion and slows down. When the speed gets low enough, the balls drop and bring the cam against the roller which causes the lever to open the gas valve.

The slide valve is not used on recent engines; it has been replaced by poppet valves. The slide valve was kept on its seat by strong springs, and there was considerable waste of power due to the friction of the valve and its two seats. An increase of pressure in the engine cylinder made it necessary to use more powerful springs. This defect made it undesirable to use high pressure; but as high pressures increase the economy, it became necessary to change the style of valve.

In the Lenoir engine the ignition was accomplished by an electric spark. In the later engines this was not used — a gas flame was substituted. The electric spark, however, is now used, as the improved mechanism makes it reliable, and this method saves considerable gas.

There have been a few improvements which are worthy of attention. The poppet valves allow a reduction in the volume of the clearance and an increase in the pressure at which the charge may be compressed before ignition. These cause a decrease in the fuel consumption. The amount of gas necessary for a slide-valve engine is about 25 per cent greater than for a poppet-valve engine. In some engines, the governor acts through the mechanism so that the opening of the admission valve is changed and the quantity of the charge reduced instead of being shut off altogether.

A Gasoline Engine is an eternal combustion engine that uses gasoline as a fuel. There is but little difference between a gas

engine and a gasoline engine. The gasoline is vaporized before it enters the cylinder by means of some device.

TESTING A GAS ENGINE.

Object of the Test. The main object of this test was to ascertain, by actual experiment, whether it is more economical

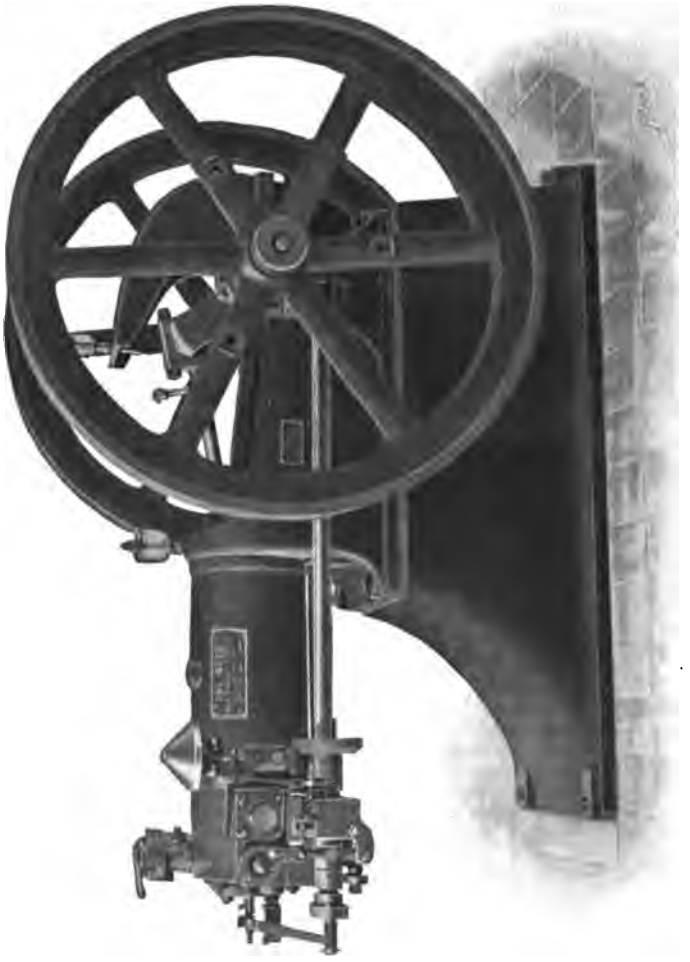


Fig. 3. The "Otto" Gas Engine.

to govern the gas engine by throttling the gas, or by omitting explosions. Gas engines may be governed by means of a governor that, when the speed increases throttles the gas and allows a weaker explosion, or the governor may regulate the valve so that

there will be fewer explosions per minute; that is, some of the explosions are missed.

The above was, however, only the main object. In addition, several important items were to be found, and deductions made from the data obtained from the test. Among these were the following:

Indicated horse-power.
Brake horse-power.
Mechanical efficiency.
Gas used per I. H. P. per hour.
Gas used per B. H. P. per hour.
Heat distribution.
Most economical mixture.

In order to determine the above-named facts, many short tests were conducted, during which data was recorded. These short runs were made with varying conditions of load, temperature of the jacket, and mixture of air and gas. To facilitate the taking of data, in fact to make it possible to obtain some of the items, various instruments and devices were attached to the engine; these devices will be described under the various headings.

Description of Engine. The engine used in this test was a small horizontal gas engine of the "Otto" type of $3\frac{1}{2}$ rated horse-power at 340 revolutions per minute. It had a stroke of 10 inches, thus giving it a normal piston speed of $\frac{340 \times 10 \times 2}{12} = 566\frac{2}{3}$ feet per minute. The piston was $4\frac{3}{4}$ inches in diameter, and hence had an area of 17.72 square inches. In order to insure steadiness, there were two fly-wheels keyed to the crank shaft. The valves and igniting apparatus were controlled by cams on a countershaft which was geared by bevel gears to make one-half as many revolutions as the main shaft. The mixture of gas and air was ignited by means of the electric spark, the current being obtained from storage batteries. On account of the high temperature in the cylinder, it was surrounded by a water jacket, the water passing from the city mains into the jacket by means of a small pipe. A valve in the pipe allowed the flow of water to be easily regulated.

INDICATED HORSE-POWER.

In order to find the indicated horse-power of a gas engine, the following must be known:

The mean effective pressure in pounds per square inch.
The area of the piston in square inches.
The number of working strokes per minute.
The length of stroke in feet.

The product of these four quantities divided by 33,000 is the I. H. P.

The mean effective pressure was found in the same way as when testing the steam engine. Cards were taken by an indicator, the area divided by the length, and the result multiplied by the scale of the spring. A Crosby indicator was used with a 120, 140, or 240-pound spring, according to the load of the friction-brake. The reducing motion was the simple contrivance shown

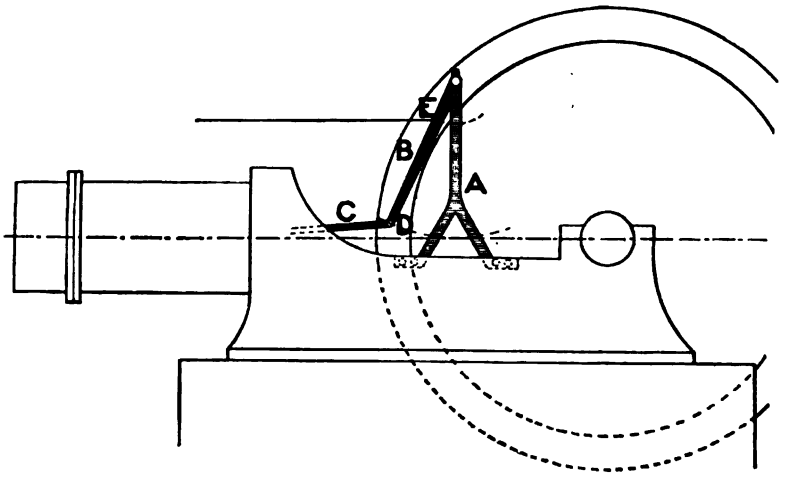


Fig. 4.

in Fig. 4. The vertical forging A was bolted to the engine frame on the right-hand side of the engine looking toward the cylinder from the crank shaft. Near the top of this support, a lever B was pivoted at one end and the other end was connected to the link C which was jointed to the wrist pin. The joint D moved in the arc of a circle as did the point E on the lever B. At this point E, the cord to the indicator drum was attached.

The Mean Effective Pressure was then found from the card.

The Length of Stroke was 10 inches or $\frac{10}{12}$ of a foot.

The Area of the Piston was 17.72 square inches. This area is simply the area of a circle $4\frac{3}{4}$ inches in diameter. It is found

by multiplying the constant .7854 (which equals $\frac{3.1416)}{4}$ by the square of the diameter.

$$.7854 \times (4.75)^2 = \text{area.}$$

$$.7854 \times 22.5625 = 17.720 \text{ square inches.}$$

The Number of Working Strokes was found by reading the speed counter attached to the admission valve. When the engine was running with no missed explosions, the number of working strokes could be verified by dividing by 2 the reading of another speed counter which was attached to the counter shaft. The reason for dividing by 2 was as follows: In the Otto engines, the cycle or events of the stroke are: Admission of air and gases during the first stroke, compression of this mixture during the return stroke, explosion followed by expansion during the next stroke, and the exhaust of the products of combustion at the fourth stroke. Hence, as the countershaft makes one-half as many revolutions as the main shaft, and the explosion occurs only at one stroke in four, the number of explosions is equal to the number of revolutions of the countershaft divided by 2.

As all four quantities are known, the I. H. P. can be calculated. Suppose the engine was making 342 revolutions per minute, and was not missing any explosions. What would be the I. H. P. if the mean effective pressure was 52 pounds per square inch?

The M. E. P. = 52 pounds per square inch,

The number of working strokes $= \frac{342}{2} = 171.$

The area of the piston = 17.72 square inches.

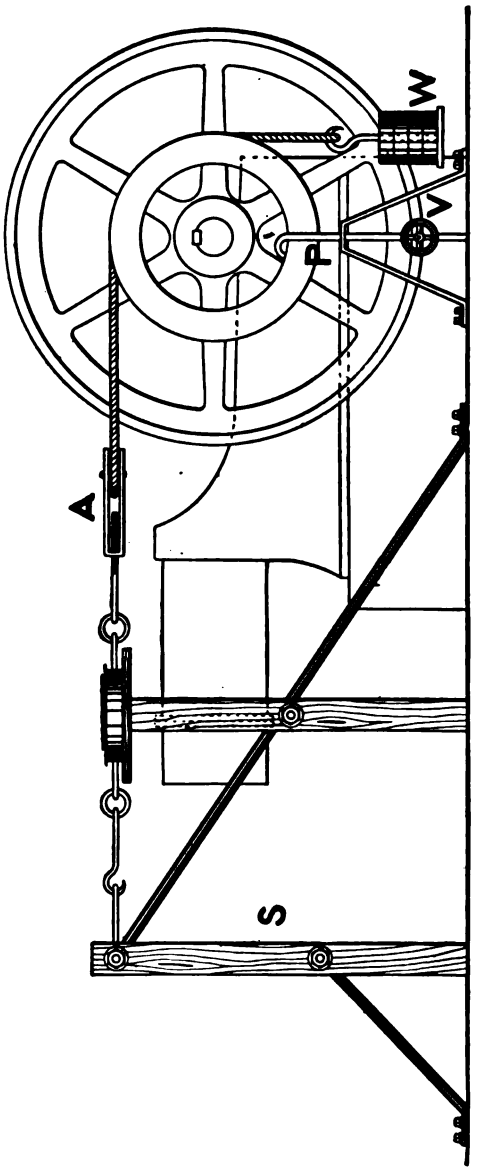
The length of stroke = $\frac{10}{12}$ feet.

Then,

$42 \times 171 \times 17.72 \times \frac{10}{12} = 131{,}305.2$ foot pounds.

and $\frac{131305.2}{33{,}000} = 3.979$ I. H. P.

In these tests it was necessary to find the I. H. P. so many times that the above calculation would be tedious. In each case 17.72 and $\frac{10}{12}$ are multiplied together and divided by 33,000. To shorten the work these three constants are formed into a single constant.



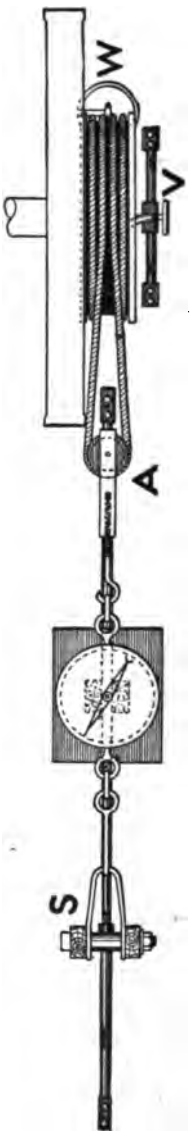


Fig. 5.

$17.72 \times \frac{19}{12} \div 33{,}000 = .0004475.$

By using this constant the I. H. P. is easily found.

To find the I. H. P., multiply the constant by the mean effective pressure and by the number of working strokes.

Thus, $.0004475 \times 52 \times 171 = 3.979.$

BRAKE HORSE-POWER.

In testing the gas engine it is desirable to know how much power is given out by the engine. The I. H. P. is the power developed in the cylinder, but it is not the net power or the power that is available for running machinery. Some power is used in running the engine; that is, some power is necessary to overcome the friction of the moving parts.

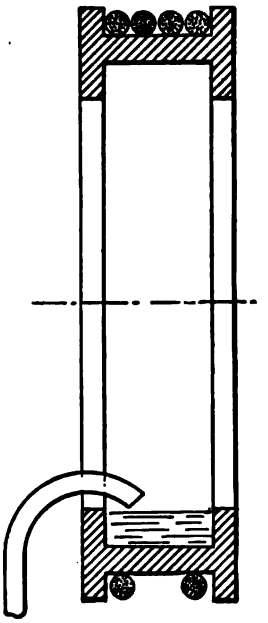


Fig. 6.

In order to find this, there must be some method used which gives either the available power or the power lost in friction. Sometimes the engine is run at the usual speed with all machinery disconnected; cards and necessary data being taken. Another method, and one that eliminates to a great extent the chances of error in the first method, is the use of the rope brake. This device absorbs all the power given out by the engine; hence if the power thus absorbed is subtracted from the I. H. P., the result will be the amount of power used in overcoming the friction of the engine.

The construction and action of the brake used in this test are easily understood by reference to Fig. 5, which shows an ordinary rope brake attached to the engine. The brake is made by winding several coils of rope around the pulley. For large powers the number of turns of rope may be increased. The pulley was shaped as shown in Fig. 6, the flanges keeping the rope in place. In case the pulley has no flanges, the rope may be held on to the face by means of blocks of wood through the grooves of which

the rope is made to pass. One end of the rope was attached to the weights W, and the other end to a spring balance. The weights were added by placing them on the holder.

There were no convenient walls or ceiling to which to fasten the spring balance, so the arrangement shown in Fig. 5 was devised. The spring balance was supported on a smooth, rectangular piece of board which was nailed to the top of a piece of $2'' \times 4''$ timber. The spring balance was securely fastened to the vertical support S which was formed of $2'' \times 4''$ lumber. The illustration shows the method of bracing by iron rods. In order that the spring balance might be free to move, and the pull be direct, the arrangement at A was adopted. A little oil or grease made the brake run smoothly and noiselessly.

As the engine ran at 350 revolutions, the pulley soon became hot; to reduce the heat a very small stream of water was allowed to flow from the city mains to the inside of the rim which was trough-shaped (see Fig. 6). The small pipe P led to the pulley, the valve V controlling the flow. On account of the shape of the rim, considerable water was carried around. A scoop could have been arranged to carry off the water, but as the amount was very small, the valve V was opened just the right amount to allow as much water to flow through as evaporated. This was easily regulated while the engine was warming up.

While the engine was being tested, the following data was recorded for the calculation of the B. H. P.; the reading of the spring balance, the weight W, and the number of revolutions per minute. To make the calculation, the difference of the readings of the weights and spring balance is multiplied by the linear velocity of a point considered as being at the center of the rope. The rope was one inch in diameter, and the pulley 16 inches in diameter. Then the radius of the circle, measured to the center of the rope, is $\frac{16}{2} + \frac{1}{2} = \frac{17}{2}$ inches, $= .7083$ feet. The circumference of this circle in feet, multiplied by the number of revolutions per minute is the linear velocity. In finding the I. H. P., the number of working strokes is one of the factors, but as the brake acts constantly, the whole number of revolutions of the pulley enters into the calculation.

Suppose the engine was running at 330 revolutions. The spring balance registered 114 pounds, and the weights were 25 pounds; what was the B. H. P?

$$\text{B. H. P.} = \frac{\pi d \times n \times (\text{spring balance} - \text{weights})}{33000}$$

$$= \frac{3.1416 \times 1.4166 \times 330 \times (114 - 25)}{33000}$$

$$= 3.96.$$

MECHANICAL EFFICIENCY.

The mechanical efficiency of any engine is the ratio of the net horse-power to the indicated horse-power, or it is $\frac{\text{B. H. P.}}{\text{I. H. P.}}$. Or, in other words, it is the ratio of the power given out at the pulley to the power developed in the cylinder. Thus, if the B. H. P. was 3.96, and the I. H. P. at the same time was 4.83, the mechanical efficiency was $\frac{3.96}{4.83} = .82$ (nearly) or 82 per cent. As the mechanical efficiency was 82 per cent, .82 of the power developed could be used.

It is always desirable to know the mechanical efficiency; it is an indication of the workmanship and design of the engine. If, during the test, the mechanical efficiency suddenly decreased, it indicated that something was wrong; a bearing getting hot on account of insufficient lubrication, piston cutting the cylinder, valve-gear out of order or something of that nature. The mechanical efficiency does not vary much with the load, as it is nearly constant under all loads. On account of the bevel gears on the countershaft, and the many parts in the valve-gear and igniting apparatus, the mechanical efficiency of a gas engine is usually less than that of a good steam engine.

GAS USED PER HOUR.

The gas used in running the engine was coal gas from the city gas mains. To the pipe leading to the engine two gas meters, whose dials indicated cubic feet, were connected, one serving as a check on the other.

Care was taken that the gas measured by the meters was

used only by the engine, and not for other purposes. As it is not easy to estimate exactly the reading at a given time, it is generally found more accurate to wait and note the time that the pointer indicates a whole number of cubic feet. This can be done accurately with but slight delay.

The number of cubic feet of gas used per hour is divided by the I. H. P. to find the number of cubic feet of gas used per I. H. P. per hour. Also, it is divided by the B. H. P. in order that the amount of gas used per net horse-power can be known.

AIR USED PER HOUR.

To determine the number of cubic feet of air used per hour, a large air meter was arranged so that all the air entering the cylinder was measured by the meter. This meter stood within six feet of the engine, making it easy to read it frequently if necessary.

HEAT DISTRIBUTION.

One of the most important items in connection with a gas engine test is the heat distribution; that is,

The proportionate amount of heat that is converted into work.
The proportionate amount that is absorbed by the jacket.
The proportionate amount that leaves at the exhaust.
The proportionate amount that is wasted.

Of course these can be determined only if the total amount of heat supplied to the engine is known. To find the heat value of a cubic foot of gas used in these tests, a volumetric analysis was obtained each day at the office of the gas company. These analyses were very similar to those given in a book by a German authority, and because of the similarity, the heat values given in the German book were used.

The heat value of one cubic foot multiplied by the number of cubic feet of gas used per hour gave the number of heat units supplied to the engine per hour.

For convenience and for comparison the heat distribution was divided into three parts.

1. B. T. U. to jacket.
2. B. T. U. to work (indicated).
3. B. T. U. to exhaust.

The last division included the heat to exhaust and all heat lost by radiation, conduction, etc.

HEAT TO JACKETS.

The amount of heat that entered the jacket water was easily found. It was only necessary to know the weight of water that flowed through the jacket per hour, and the number of degrees Fahrenheit that the water was raised in temperature. To find the weight of water, a small pipe was connected to the jacket; the end emptied into a barrel which stood on platform scales. A thermometer graduated to 212° F. was inserted in this pipe close to the engine. The pipe was vertical, and by means of a right-angle bend and a plug, the thermometer could be inserted so that all the water from the jacket would flow around the bulb. The temperature of the water as it entered the jacket was of course the same as that in the city mains. As this temperature was practically constant for several hours, the temperature was taken once about the middle of the trial.

A British thermal unit is the amount of heat necessary to raise the temperature of one pound of water one degree Fahrenheit. Then if the number of pounds of water used per hour is multiplied by the number of degrees that the water is warmed, the result will be the number of heat units given to the water per hour.

Suppose 250 pounds of water ran into the barrel; the temperature was 55 degrees in the mains and 170 in the jacket. Then 250 pounds of water were raised $170 - 55 = 115$ degrees. The B. T. U. to the jacket was therefore $115 \times 250 = 28{,}750$ per hour.

HEAT UNITS TO WORK.

As in the case of the amount of heat absorbed by the jacket water, the amount of heat converted into work was easily calculated. Scientific experiments show that one heat unit (B. T. U.) is equivalent to 778 foot-pounds of work. This constant is often called the mechanical equivalent of heat. Then if the number of foot-pounds of work done per hour is known, the B. T. U. per hour is that quantity divided by 778. For example, during run No. 9 (see Fig. 8) the I. H. P. was 3.76, which equals 3.76 ×

33,000 = 124,080 foot-pounds of work per minute. The work done per hour is 124,080 × 60 = 7,444,800. The B. T. U. per hour equals $\frac{7,444,800}{778} = 9,569$.

A somewhat shorter method, especially if the result is to be computed many times, is to first find the B. T. U. per I. H. P. or $\frac{33,000 \times 60}{778} = 2,545$. Then the B. T. U. for 3.76 I. H. P. is 2,545 × 3.76 = 9,569.

B. T. U. TO EXHAUST.

It is readily seen from the distribution of heat as already stated, that the heat to exhaust and heat lost must be the amount remaining after the B. T. U. to the jacket and B. T. U. to work have been subtracted from the total heat. Thus, suppose the total heat per hour was 9,683 B. T. U., and the sum of the amounts to jacket and work was 6,742 B. T. U.; the heat to exhaust and unaccounted for was 9,683 — 6,742 = 2,941 B. T. U.

The heat distribution was easily reduced to per cent by considering the total heat supplied as 100 per cent and finding what proportion was converted into work, absorbed by the jacket, etc.

Thus, 10,114 B. T. U. was supplied per hour; 4,960 B. T. U. was absorbed by the jacket water; 2,112 was converted into work, and 3,042 to exhaust and unaccounted for.

Then,

$$\frac{4,960}{10,114} = .49 = 49 \text{ per cent to jacket.}$$

$$\frac{2,112}{10,114} = .208 = 20.8 \text{ per cent to work.}$$

$$\frac{3,042}{10,114} = .30 = 30 \text{ per cent to exhaust.}$$

MAKING THE TEST.

About three hours were usually allowed for making a test; this gave one to two hours to get the engine running steadily under the desired conditions, and about the same time for taking data.

On commencing a test, there was certain preliminary work to

do. The thermometers had to be inserted in the pipes from the exhaust and from the jacket, the indicator attached, and the barrel on the scales partially filled with cold water. In addition, it was necessary to inspect all parts of the brake, the speed counters, and the air and gas meters. The engine was then carefully oiled, the connections made to the storage batteries, and the jacket-water turned on. The engine was then started. For about an hour the engine was allowed to run while adjusting the supply of water to the jacket so that the temperature would be constant at about 160° to 170°. This required some care, because the cylinder would be injured if the temperature of the water becomes too high. If the water in the jacket were converted into steam, the cylinder would be in danger, since steam is a poor conductor of heat. A few cards were taken at intervals to see how the engine was running, and to see if the indicator and rig were in good working order. The gas valve and the weights on the brake were adjusted so that the engine would run at about 340 revolutions per minute. Mental note was made of the various data, and after the engine became thoroughly heated and everything was running smoothly and constantly, the necessary data was recorded. The following operations were necessary for these tests:

Taking indicator cards every five minutes.

Weighing the jacket water every ten minutes.

Reading the thermometers and the spring balance every five minutes.

Reading two speed counters every five minutes.

Reading the gas and air meters about every ten minutes.

Between readings a graphical log. was made on co-ordinate paper. One of these logs with the lines named and the quantities attached is shown in Fig. 7. For these logs, time was plotted as abscissæ, as seen at the bottom of Fig. 7, and the following quantities as ordinates; revolutions per minute, explosions per minute, cubic feet of gas, cubic feet of air, jacket temperature and exhaust temperature. By means of this log it is easy to see if there are any variations in the quantities recorded.

These logs besides showing the steadiness with which the engine is running, are, with the indicator cards, a means of preserving data.

THE SERIES OF RUNS.

The first series of runs, numbering 15 in all, was made while governing by throttling the gas; that is, no explosions were missed. In this series, great care was necessary in making the runs because any slight increase in load, due to lack of lubrication or other

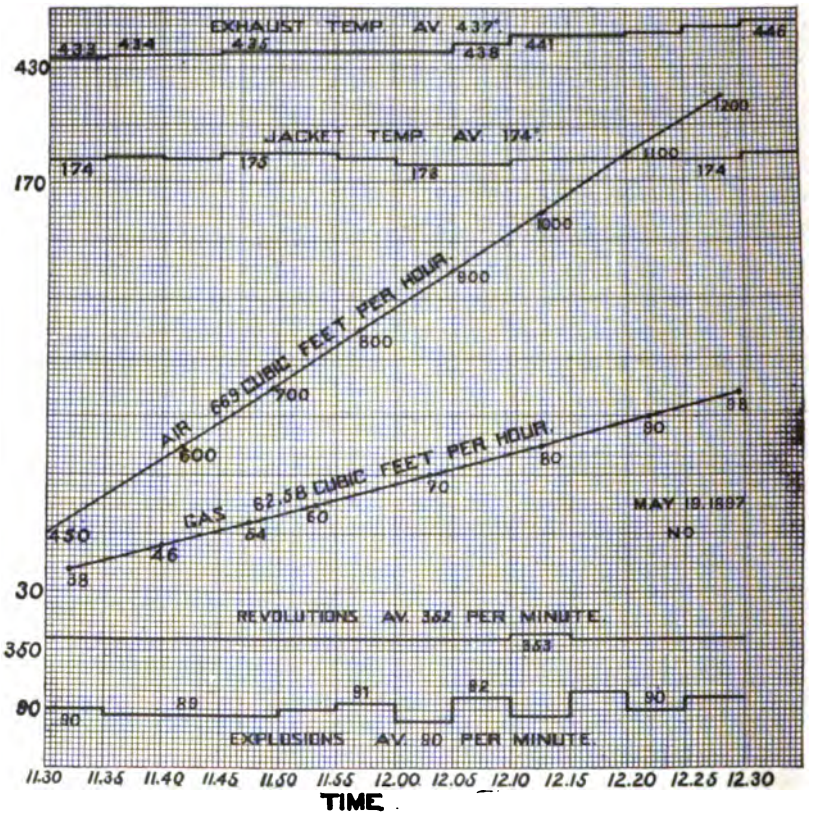

EXHAUST TEMP. AV. 437°
JACKET TEMP. AV. 174°
AIR 669 CUBIC FEET PER HOUR
GAS 62.58 CUBIC FEET PER HOUR
MAY 19.1887
NO
REVOLUTIONS AV. 362 PER MINUTE
EXPLOSIONS AV. 90 PER MINUTE
TIME
11.30 11.35 11.40 11.45 11.50 11.55 12.00 12.05 12.10 12.15 12.20 12.25 12.30

Fig. 7.

cause, would make the engine slow down. The reason for this was that just the right amount of gas and air was admitted to do the work, and if the work increased, there being no additional supply of gas, the number of revolutions decreased. As there was no attachment on the engine for governing by this method, the load was fixed, and the gas-valve adjusted to admit the proper amount of gas.

These runs were made under varying conditions of load, and the series was continued until the number of runs should furnish sufficient data for a reliable curve.

After the first series was completed, the second series was begun. In these runs, the loads were varied as before, but now the gas-valve was opened so that plenty of gas would be admitted; the governor regulating the speed by causing some explosions to be admitted. The manner of taking data, and the calculations made were the same as for the first series.

The second series was subdivided, the gas-valve being set at a certain position during the whole set.

After making the tests, or at any spare time during the tests, the data was worked into definite shape. The calculations, as previously explained, were made from the indicator cards, and the notebooks. In order to preserve the data aud results, it was tabulated in the form shown in Fig. 8. The columns at the left being mainly data, and those at the right, results. The heavy horizontal line after the fifteenth run separates the first series from the second; the remaining heavy lines in the second series indicate the subdivisions caused by the positions of the gas-valve.

PLOTTED CURVES.

By referring to the column headed "Cubic Feet of Gas per Hour per B. H. P." (Fig. 8), it is seen that the values in this column are less for the first series than for the second. This indicates that less gas was used per B. H. P. when the engine was governed by throttling the gas, than when governed by missing explosions. In other words, the test showed which was the more economical means of governing that engine.

In order to see graphically the result of the test, the curve shown in Fig. 9 was plotted. To do this, the brake horse-powers were plotted as abscissæ, and the number of cubic feet of gas per brake horse-power per hour as ordinates. The points thus determined for the first series were marked with a circle, and those of the second series marked with a cross. The curves thus drawn show graphically that in general the points for the first series were lower than for the second, indicating that the throttling was the more economical method. The curves for the two series approach

each other as the power increases. This is due to the fact that as the power increased, fewer explosions were missed, and the conditions approached those of the first series.

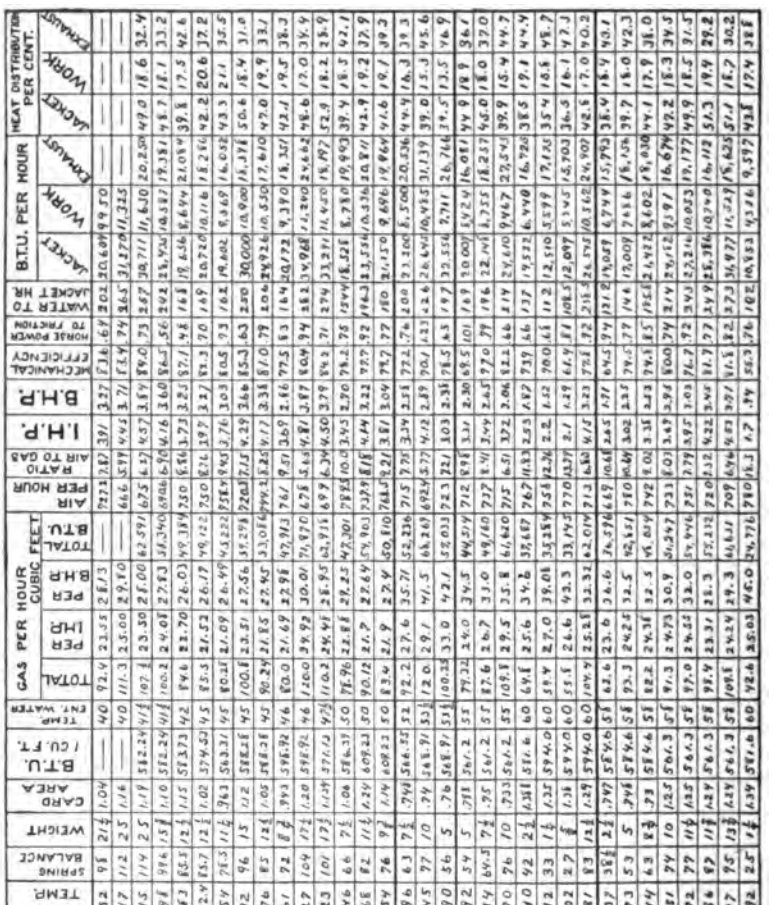
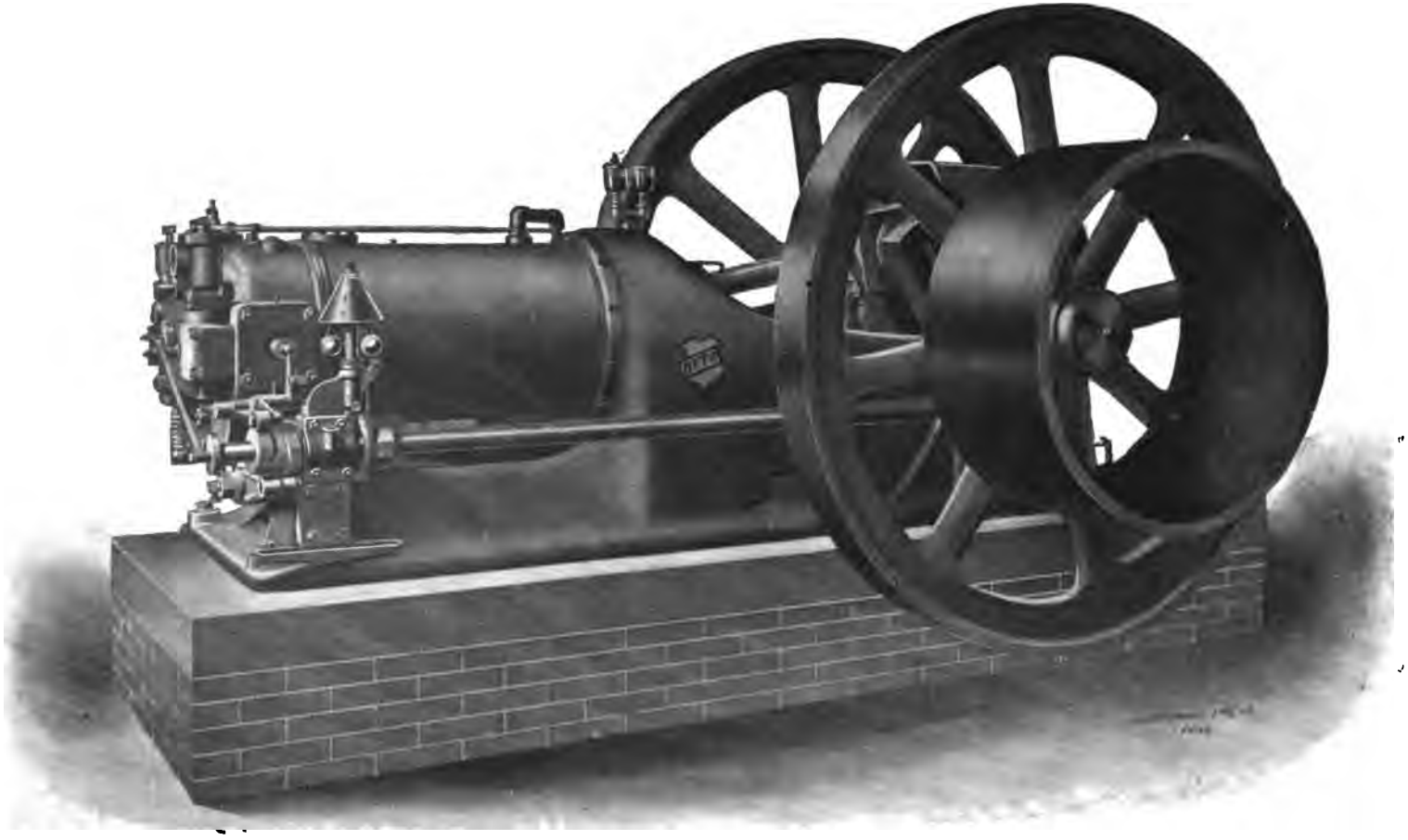


Fig. 8.

The curve also showed that for light loads the consumption of gas is high; in other words, it was not economical to run at about one-half or one-third the rated power.



THE OTTO GAS ENGINE.

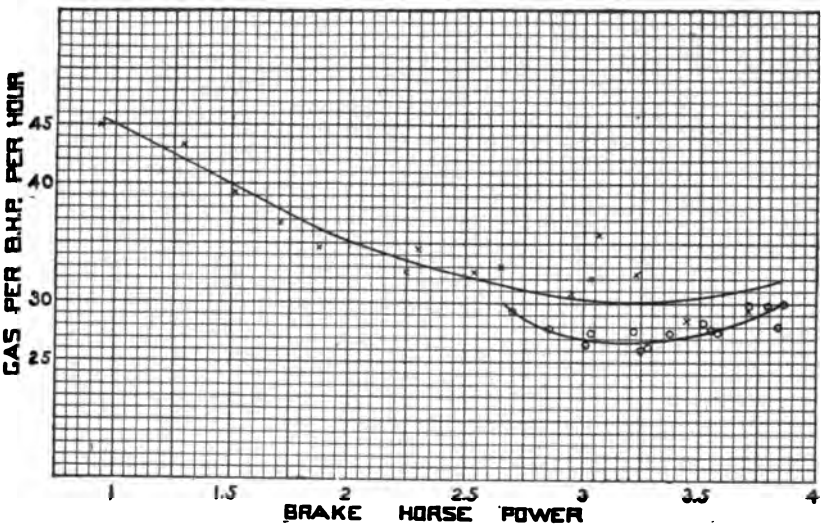

GAS PER B.H.P. PER HOUR
45
40
30
25
1
1.5
2
2.5
3
3.5
4
BRAKE HORSE POWER

Fig. 9.

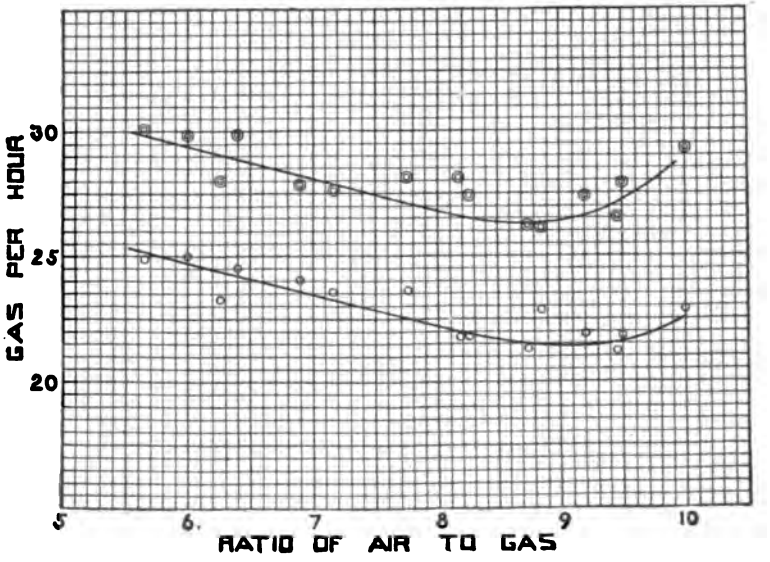

GAS PER HOUR
30
25
20
5
6
7
8
9
10
RATIO OF AIR TO GAS

Fig. 10.

The most economical net power was about 3.25 horse-power, as shown by the curve; the lowest points of the curve corresponding to this power.

The reason why there were no points for light loads in the first series was that unless the power developed was considerable, the engine would not keep up to speed unless some explosions were missed.

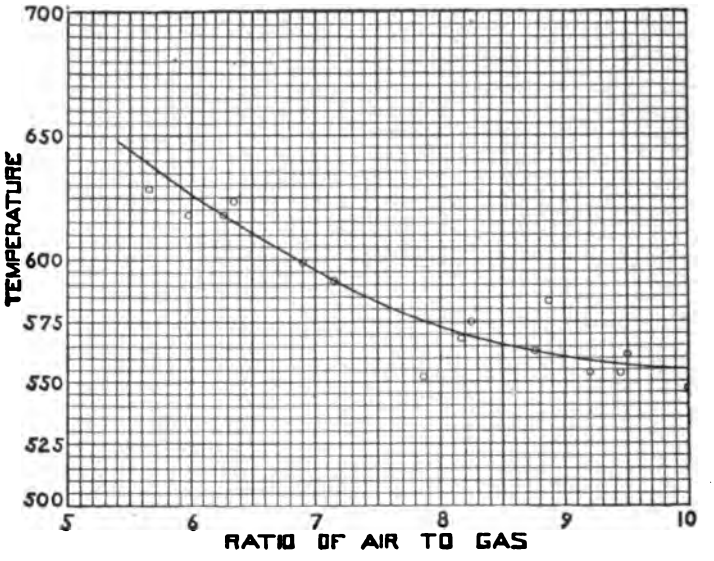


Fig. 11.

The curve shown in Fig. 10 was next plotted. For abscissæ, ratios of cubic feet of air to cubic feet of gas were taken. Cubic feet of gas per hour per horse-power (both brake and indicated) were plotted as ordinates. The points having cubic feet per brake horse-power are indicated by the double circle. These two curves are nearly parallel, showing that the mechanical efficiency was about constant. The curves also show that the most economical mixture is the ratio of about 1 : 9.

Points were located which were the results of the first series of runs. In these runs the conditions were more uniform, and the engine was working under average loads.

A third curve, Fig. 11, was plotted, with ratios of cubic feet

of air to cubic feet of gas as abscissæ and exhaust temperatures as ordinates. The curve shows a decided increase of temperature as the mixture was strengthened, or as the ratio became less.

The temperature of exhaust is an indication of the amount of heat wasted in the exhaust, hence the lowest temperature should correspond to the most economical mixtures. The curve shows that the lowest temperature corresponds to a ratio of about 1:9. The curve of Fig. 10 also shows 1:9 to be the most economical mixture. For this curve (Fig. 11) only points obtained from the runs of the first series were used, because when missing explosions the temperature of exhaust varies with the number of missed explosions, and besides the first series furnished plenty of points.

INDICATOR CARDS.

From the large number of indicator cards taken during these

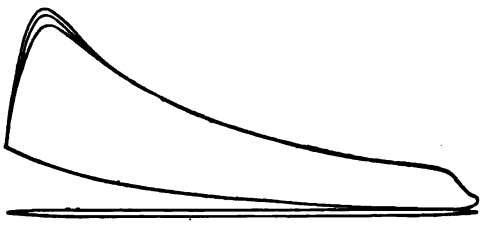


Fig. 12.

tests, four are shown here as typical. The card shown in Fig. 12 was an average card. It has no especial features, but is one that was taken frequently when the engine was running at a moderate power with a fairly strong mixture. It is typical for average performances.

The diagram shown in Fig. 13 is what was called a very good card; in fact, it was perhaps the best that was taken. The nearly vertical line at the left end with the high peak coming almost to a point, denotes a strong mixture and a sharp, quick explosion. The expansion line is smooth, which shows that there was little or no prolongation of the explosion, or no after-burning. The rest of the card is similar to the corresponding parts of the other cards.

Fig. 14 shows a poor card. The vertical line which should be at the left is missing, and in its place is a jumble of curved lines. The low curves having no pronounced peak, indicate a slow ignition, slow explosion, and a weak mixture. Cards like Fig. 14 were obtained only when the mixture was weak.

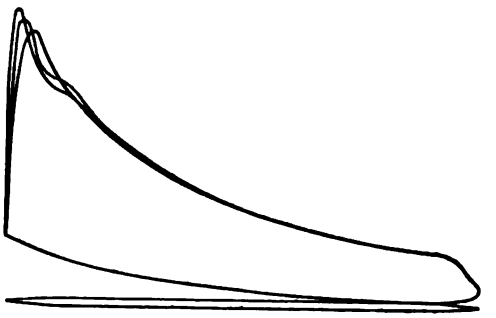


Fig. 13.

The card shown in Fig. 15 has a good vertical line at the left, and a high peak which denotes a high temperature at that point. The expansion curve is, however, wavy, showing either vibration

Fig. 14.

of the indicator spring or after-burning; that is, all the air and gas did not explode it at once, but continued to burn as the piston moved forward.

In conclusion, the results of the tests may be stated briefly as follows:

It was found that for a small engine it is more economical to govern the speed by throttling the gas than by missing explosions. Also the power obtained was more nearly constant when the engine was governed the more economical way. The amount of heat converted into work was equal to from 18 to 20 per cent of

the total heat of the gas supplied to the engine. The heat to the jacket was from 40 to 50 per cent. The most economical mixture was about 1:9; and the most economical power for the engine was about 3.25 horse-power.

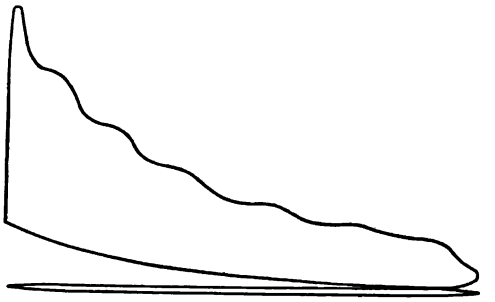


Fig. 15.

APPENDIX

QUESTIONS IN REVIEW

The following lists of questions afford the reader a means of testing his knowledge of the subjects treated in the present volume. Any one who has carefully read the foregoing subjects should be able to answer every question accurately. The reader is advised to go through the questions thoroughly, working out his answers and checking his results with the answers given. This will afford a valuable means of fixing the matter in mind. Students who are preparing for examinations for college, civil service or engineer's license will find the questions of the greatest value.

REVIEW QUESTIONS

ON THE SUBJECT OF

MACHINE DESIGN.

---

1. A wheel 42 inches in diameter makes 35 revolutions per minute. What is the linear velocity of a point on the circumference? Ans. 6.4 feet per second.

2. If the angular velocity of a wheel 6 feet in diameter is 30 per second, what is the linear velocity? Ans. 90 feet per second.

3. Define reciprocating motion, continuous motion, and intermittent motion.

4. What is a machine?

5. Why are the relative movements of the parts of a machine independent both of the power transmitted and the size of the parts?

6. Define power and its unit.

7. Why are not foot-pounds an indication of horse-power?

8. Name some forms in which potential energy may exist.

9. What is the energy of a body weighing 20 pounds and moving with a velocity of 4 feet per second?

Ans. 4.97 foot-pounds.

10. In designing an engine why would you make the piston rod of wrought iron or steel rather than of cast iron?

11. Explain with sketch why all corners of castings should be well rounded.

12. Why are castings weak if some parts are much heavier than others?

13. What elements and processes are employed to get sound steel castings?

14. Name the principal strains in machines and the methods used in designing to allow for these strains.

15. What metals are used for bearing alloys?

16. If you were making a working drawing of an engine crank, how many and what views would you make?

17. When are sectional views used?

18. What is the difference between brass and bronze? Which of these two alloys is more frequently used for bearings?

19. Name some part of a machine that is designed from calculation for strength. Some part designed to provide for wear.

20. Describe with sketch the standard U. S. screw or bolt thread.

21. What is the stress per square inch on a bolt if the diameter is $1\frac{1}{4}$ inches and the load 3,800 pounds?

22. When is a screw made with multiple threads in preference to a single thread? Why?

23. Describe the standard bolt and nut.

24. Why is the Sellers thread not suited for gas pipe?

25. An engine cylinder head is bolted to the cylinder with 8 bolts. If the maximum total steam pressure on the cylinder head is 28,000 pounds, what should be the diameter of the bolts? Assume safe working stress as 5,000 pounds.

Ans. $1\frac{1}{8}$ inches in diameter.

26. What is the pitch of a screw? Make your definition applicable to either single or multiple threaded screws.

27. If a standard bolt is $1\frac{5}{8}$ inches in external diameter, what is the pitch? Find by formula. Ans. .185 inch.

28. Describe the Whitworth thread.

29. (*a*) What are the special advantages of taper threads compared with square threads? (*b*) For what are buttress threads used?

30. Describe with sketch some method of fastening foundation bolts.

31. What is a key?

32. Describe the knuckle joint. Name some common form of this joint.

33. Is the Grover spring a good locking device? Why?

34. Describe with sketch what you consider a good locking arrangement for nuts.

35. Why are pin keys used on small shafts in preference to rectangular keys? Mention any disadvantage they possess.

36. Describe a method of fastening small engine cranks to shafts.

37. Why are keys tapered? About how much is the taper?

38. What is the most effective form of key? Why?

39. Find the length of a steel key for a 3-inch shaft which transmits 50 horse-power at 100 revolutions per minute. Make the width of the key 1 inch. Ans. 3 inches.

40. Find the dimensions of a cotter and rod of the form shown in Fig. 42. The load is 5,500 pounds and $S_t = 7{,}000$ pounds.

Ans. $\begin{cases} d = 1\frac{1}{4} \text{ inches} \\ d_1 = 1 \text{ inch} \\ d_2 = 1\frac{7}{16} \text{ inches} \\ D = 2\frac{1}{2} \text{ inches} \\ t = \frac{5}{16} \text{ inch} \\ b = 1\frac{9}{16} \text{ inches} \\ a = 1\frac{5}{16} \text{ to } 1\frac{9}{16} \text{ inches.} \end{cases}$

41. Make a sketch of a locking device for a cotter that has considerable taper.

42. A cotter is 2 inches wide at one end and $2\frac{7}{8}$ at the other. If it is 14 inches long, what is the taper per foot?

43. Find the proper length and diameter of a steel end journal for a load of 2,300 pounds. Assume $\frac{l}{d} = 1.75$, and the safe working stress S, as 8,500 pounds.

Ans. $1\frac{9}{16}$ in. diameter and $2\frac{3}{4}$ in. long.

44. Find the proportions of a steel journal for a load of 10,000 pounds. Assume a safe working stress of 9,000 pounds, and an allowable pressure of 850 pounds. Ans. $2\frac{7}{8} \times 4\frac{1}{8}$ inches.

45. Find the diameter and length of a wrought iron neck journal when $W = 5{,}000$, $S = 7{,}500$, and $\frac{l}{d} = 2$.

Ans. $1\frac{5}{16}$ in. diameter and $2\frac{5}{8}$ in. long.

46. What should be the height and breadth of collar for the above journal? Ans. $\frac{1}{4}$ in. in height and $\frac{3}{8}$ in. in breadth.

47. What should be the diameter of a pivot journal for a load of 25,000 pounds and an allowable pressure of 600 pounds per square inch? Ans. $7\frac{1}{4}$ inches.

48. The end thrust on a 9-inch shaft is 12,000 pounds. Find the diameter of the collars, assuming 5 are used.

Ans. D = $11\frac{1}{2}$ inches.

49. What should be the diameter of a mill shaft to transmit 55 horse-power at 90 revolutions per minute? Ans. $3\frac{13}{16}$ inches.

50. Why is a hollow shaft stronger than a solid one of equal weight? Explain fully.

51. Find the outside and inside diameters of a hollow shaft that equals in strength a solid shaft 8 inches in diameter. The inside diameter to be $\frac{1}{2}$ the outside. Ans. $8\frac{3}{16}$ and $4\frac{3}{32}$ inches.

52. What limits the distance between bearings on a line of shafting?

53. Describe with sketch a good simple shaft coupling.

REVIEW QUESTIONS

ON THE SUBJECT OF

MACHINE DESIGN.

---

1. A wheel 40 inches in diameter, faced with paper, drives a cast iron wheel at 140 revolutions per minute. If the force pressing the wheels together is 550 pounds, what is the horse-power transmitted? Ans. 4.89 H. P.

2. Why are bearings divided?

3. How is the relative speed of the surfaces in contact of pivots reduced?

4. Sketch a footstep bearing for a shaft 2 inches in diameter.

5. What is a gasket?

6. What is babbitting?

7. What metals are used to line bearings?

8. How should a bearing be divided?

9. What curves are used for the faces of gear teeth?

10. Why are curves instead of straight lines used for the faces and flanks of teeth?

11. A pivot runs at 2,250 revolutions per minute; if 4 discs are used, what is the relative speed of adjacent surfaces?

Ans. 450 revolutions per minute.

12. In the Sellars bearing, in what manner is it constructed to be self-adjusting? How is overheating prevented?

13. What is circular pitch?

14. What is a cycloidal curve?

15. How is vertical adjustment obtained in the hanger shown in Fig. 15?

16. Define addendum, clearance, backlash, and dedendum.

17. What is the pitch circle? Explain what it represents, and how it is used in designing.

18. How is lateral and vertical adjustment obtained in foot-step bearings?

19. Why cannot large powers be transmitted by friction gearing? Explain in full.

20. Sketch the section of a rim of a gear wheel having teeth shrouded in such a manner that but one of the pair can be shrouded.

21. A spur wheel has 38 teeth and is 19 inches in diameter. Find the diametral and circular pitches.

Ans. $\begin{cases} D = 2. \\ P = 1.571 \text{ inches.} \end{cases}$

22. A gear is 12 inches in diameter. If the circular pitch is .785, how many teeth has it?

23. A pinion 8 inches in diameter has 12 teeth. What is the circular pitch? Ans. 2.094 inches.

24. If the circular pitch of a gear wheel is 1.257, what should be the thickness of the rim? Ans. $\frac{5}{8}$ inch.

25. A cast iron gear wheel is to transmit 12 horse-power at 75 revolutions per minute. If it is 30 inches in diameter, what should be the pitch and the number of teeth? Assume S = 4,000.

Ans. $\begin{cases} P = 1.047 \text{ inches.} \\ 90 \text{ teeth.} \end{cases}$

26. If a worm wheel has 64 teeth, how many turns must the worm make per minute if the wheel is to make 30?

27. A spur gear 28 inches in diameter has 6 arms. If the rim is 3 inches wide, and the diametral pitch is $2\frac{3}{4}$, find the dimensions of the arms.

Ans. $\begin{cases} \text{Width of arm} = 2\frac{3}{8} \text{ inches.} \\ \text{Thickness of arm} = \frac{5}{8} \text{ inch.} \end{cases}$

28. A worm gear has the following proportions: radius of worm, $1\frac{3}{4}$ inches; number of convolutions of thread on worm, 3; circular pitch of wheel, .898 inch. Calculate the efficiency.

29. Design a spur gear to transmit 20 horse-power at 60 revolutions per minute. Assume S = 4,000 pounds and the pitch circle 30 inches in diameter.

Ans.

Circular pitch	= 1.571 inches
Diametral pitch	= 2
Number of teeth	= 60
Thickness of rim	= $\frac{3}{4}$ inch
Breadth of rim	= 4 inches
Number of arms	= 6
Width of arm at hub	= $3\frac{3}{16}$ inches
Thickness of arm	= $\frac{3}{4}$ inch
Thickness of rib	= $\frac{1}{2}$ inch
Diameter of shaft	= $3\frac{1}{2}$ inches
Thickness of hub	= $1\frac{1}{2}$ inches
Length of hub	= 4 inches

30. A certain leather belt is 11 inches wide. What width of rim should the pulleys have which carry it?

31. Which is preferable, to arrange belts so that they will run in one direction only or in either direction, and why?

32. What effect does centrifugal force have on the friction between the rope and pulley?

33. When a rope-belt stretches with use, how may the extra length be taken up?

34. What is the principal objection to the use of chain gearing?

35. Make a sketch of pulley rims for manilla rope and for wire rope showing position of rope upon the groove.

36. What is a belt shipper and for what is it used?

37. What are some of the advantages of rope gearing over belt gearing?

38. About what is the maximum velocity at which ropes and belts should be run?

39. Of what material are ropes usually made?

40. Determine by use of the diagram (Fig. 53) what horse-power may be transmitted by a 2-inch manilla rope running at a velocity of 20 feet per second.

41. What is a ratchet wheel?

42. Why is it not advisable to undertake to transmit a large amount of power by a quarter-turn belt?

43. Why should the length of the links of crane chains be as small as possible? At what part of the link is the greatest strain?

44. What is a reversible pawl? In what machines is it used?

45. Why is it well to use a guide pulley with a quarter-turn belt?

46. Calculate the working strength for a manilla transmission rope $1\frac{1}{4}$ inches in diameter.

47. Describe the ordinary method for transmitting power long distances by wire-rope gearing.

48. In what machines, and for what purposes, are ratchet wheels used?

49. What is a chain drum?

REVIEW QUESTIONS

ON THE SUBJECT OF

CONSTRUCTION OF BOILERS.

1. If they are near together how are two flat parallel surfaces stayed?

2. Describe a rivet.

3. Since copper is such a desirable metal for boiler work why is it not used more extensively?

4. Why is a large factor of safety used for stays?

5. State what you can (briefly) about the injuries done to plates by punching and the methods employed to overcome them.

6. Why are not welded joints used more generally?

7. In what two ways are tubes fastened to the tube sheet?

8. About what is the ratio of length to diameter of the multitubular type of boiler?

9. Explain riveting with countersunk head.

10. Is the greatest tendency to rupture along the longitudinal or the circumferential seams?

11. Why is the length of a grate limited?

12. Which is the stronger form of riveting, the lap joint or double butt joint (both with double riveting)?

13. Why are the short screw stay bolts turned smooth in the center?

14. Why are flanged joints preferable to those made with cast iron angle irons?

15. What is the water-leg?

16. For what qualities are boiler materials tested?

17. What is the principal advantage of pneumatic calking?

18. Name some of the considerations that must be kept in mind when designing a boiler?

19. If the rivet is 1 inch in diameter what would *you* make the pitch, if single riveted lap joint?

20. Upon what does the choice of type depend?

21. What is the thickness of the shell of a boiler if the diameter is 60 inches, the steam pressure 70 pounds per square inch, the joint has an efficiency of .68 and the working strength of the metal 9000 pounds per square inch? The working strength equals the breaking strength S, divided by f.

Use the formula $t = \frac{fDp}{2SE}$. Ans. $\frac{3}{8}$ inches.

22. Find the allowable pressure, if the above conditions are the same, with the exception of t, which is $\frac{1}{2}$ inch.

Ans. 102 pounds per square inch.

23. Why should calking be done carefully and with a proper shaped tool?

24. About what fraction of the volume of a multitubular boiler is the steam space?

25. If the boiler is to supply steam to a high speed engine, should the steam space be larger or smaller than if to a low speed engine?

26. Why is wrought iron preferable to cast iron for boiler shells?

27. Why is a riveted joint weaker than an uncut plate?

28. A boiler evaporates 3211 pounds of water per hour. The grate is $6\frac{1}{2}$ feet by 4 feet and the coal evaporates $9\frac{1}{2}$ pounds of water per pound. What is the rate of combustion?

Ans. 13 pounds per sq. ft. per hour.

29. What kind of joint is generally used for longitudinal seams?

30. What should be the diameter at the top of a chimney 81 feet high if 2.88 pounds of coal are burned per horse-power per hour, and the horse-power is 160?

Use formula foot page 56. Ans. 28 inches.

31. Name four general requirements that a boiler must have.

32. What are the advantages of machine riveting.

REVIEW QUESTIONS

ON THE SUBJECT OF

TYPES OF BOILERS.

---

1. How was the heating surface of the Haystack boiler increased by Smeaton?

2. What kind of draft was used in Watt's Wagon boiler?

3. Under what circumstances are plain cylindrical boilers used?

4. What two important improvements were made in the plain cylindrical boiler?

5. In what ways are the internal flues of the Cornish boiler strengthened?

6. What is the principal difference between the Cornish and Lancashire boiler?

7. What is a Breeches flued boiler?

8. Describe briefly the Galloway tube and its advantages.

9. Describe briefly the Multitubular boiler.

10. Under what circumstances are vertical boilers used?

11. Why does the Cochrane boiler require little staying?

12. Give some requisites of a fire engine boiler. How is the forced draft obtained?

13. Why does the locomotive boiler, when used as a stationary boiler, require but little brick setting?

14. State the principal difference between fire-tube and water-tube boilers.

15. Give some of the advantages claimed for water-tube boilers.

16. Is the Babcock and Wilcox boiler a sectional boiler?

17. Trace the circulation of water and of the hot gases in the Babcock and Wilcox boiler.

18. Give some differences between the Babcock and Wilcox boiler and the Heine.

19. Describe briefly the peculiarities of the Harrison boiler. What are the principal advantages?

20. Mention some of the peculiarities of the Hazelton boiler.

21. How does the Cahall boiler differ from the other water-tube boilers?

22. Why was the form of marine boiler changed from box shaped to cylindrical?

23. What is the advantage of using double ended marine boilers?

24. Under what conditions is the Through tube marine boiler used? How does it differ from the Scotch drum boiler?

25. Name some water-tube boilers that are used for marine work.

26. Why is the pressure liable to vary in water-tube boilers?

REVIEW QUESTIONS

ON THE SUBJECT OF

BOILER ACCESSORIES.

---

1. When are check valves used? Explain the action.
2. What is the unit of boiler horse-power?
3. What devices are used to drain condensation from steam pipes?
4. What are the defects of the lever safety valve?
5. Why should the feed supply be regulated so that the water level shall be as nearly stationary as possible?
6. Describe the principle upon which the injector or inspirator works.
7. Why is it necessary to reduce the actual evaporation to an equivalent evaporation, before comparisons can be made?
8. Name some methods for preventing smoke.
9. Describe the two types of feed water heaters.
10. Of what are fusible plugs made? Where are they placed?
11. What are the requisites of a good boiler setting?
12. What portions of a boiler should be kept clean?
13. What is a good average heat value for coal?
14. In building a fire 360 pounds of wood were used. What is the coal equivalent?
15. State some reasons why the efficiency of a boiler and furnace is considerably less than 100 per cent.
16. Describe the pop safety valve.
17. What should be the diameter of a safety valve when

the boiler evaporates 1.846 pounds of water per second, if the lift is .1 of an inch and the pressure 100 pounds (absolute)?

Ans. 8 inches.

18. Describe the process of banking a fire.

19. What harm is caused by the admission of engine oil to the boiler?

20. In general, how are boiler explosions prevented?

21. What are the three kinds of firing?

22. Name and state the objects of the two kinds of boiler tests.

23. Describe the action of the down draft furnace.

24. Where is the feed water generally introduced?

25. Name some good pipe coverings.

26. Why is scale undesirable?

27. Why should the steam space never be exposed to the heat of the fire?

28. A lever safety valve is set to blow off at 65 pounds. The ball at the end weighs 110 pounds, the lever weighs 18 pounds and has its center of gravity 18 inches from the fulcrum. The valve is $4\frac{1}{2}$ inches from the fulcrum. The valve has a diameter of $4\frac{1}{2}$ inches and with the spindle weighs 14 pounds. At what distance from the fulcrum must the ball be placed?

Ans. $33\frac{7}{8}$ inches (nearly).

29. Describe in detail the methods for starting a fire in a boiler furnace, for bituminous coal.

30. Why are patches of scale more harmful than a uniform coating?

31. If you were in charge of a boiler and the gage cocks gave no indication of water, what would you do?

32. Describe briefly the two most important kinds of coal.

33. Show why it is economical to introduce the feed water at a high temperature. What per cent. is gained if the feed water enters at 200° F. instead of 60° F.? The pressure is 110 pounds (gage).

34. Describe the action and principle of the Bourdon gage. (Make sketch).

35. Before opening the drafts, what should a fireman do?

36. Describe briefly the different kinds of forced draft and when used.

37. Explain the importance of gage cocks.

38. Do you think mismanagement causes many boiler explosions? Why?

39. What data should be taken at a boiler trial, and at what intervals?

40. Find the diameter of the steam pipe when the total amount of steam furnished per minute is 200 pounds; the pressure is 105 pounds absolute, and the allowable velocity 5,500 feet per minute. Ans. $5\frac{1}{4}+$ inches. Use 6-inch pipe.

41. Name some general causes for boiler explosions.

42. Why does the application of high pressure injure a boiler?

43. Name three ways to prevent and remove scale.

44. Describe boiler testing by the Standard method.

45. Why is there a large amount of energy in boilers? Is it in the water or in the steam?

46. Name the principal apparatus used in boiler trials and describe their uses.

47. What impurities in feed water cause corrosion?

48. Explain the action of soda on feed water impurities.

49. The following results were obtained from a separator calorimeter test; weight of dry steam 12 pounds, weight of water separated 8 ounces, radiation loss 2 ounces. Find the quality of the steam. Ans. 3 per cent. moisture.

50. The data from a throttling calorimeter test are as follows: steam pressure 85 pounds (absolute), pressure in calorimeter 25 lbs. (absolute), and temperature in calorimeter 266.5°F. Find the quality of the steam. Ans. 1.18 per cent. moisture.

51. What is the action of kerosene oil in a boiler?

52. How is forced draft measured?

53. Find the equivalent evaporation when a boiler evaporates 1,900 pounds of water per hour; the gage pressure being 115 pounds, and the temperature of feed water 80°F. What is the horse-power of the boiler?

Ans. Equivalent evaporation 2,242 pounds. 65 H. P.

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